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Automatic Transmissions

PERHAPS the most important question that faces the automobile industry at the present time is the desirability or otherwise of adopting some form of automatic or semi-automatic transmission. This question has already been answered in the U.S.A. where a variety of such transmissions are offered as optional extras on a wide range of cars. They are, in fact, standard equipment on the more expensive models in the range of several manufacturers. Among the earlier examples were the Chrysler M.6 hydraulically operated transmission of the four-speed, constant mesh, layshaft type, and the General Motors Hydramatic, embodying a fluid coupling and three planetary assemblies providing four forward speeds. This transmission has found a wide field of application among other makers, but although it is still fitted on certain cars in the General Motors range, it would appear that it will ultimately be superseded by later developments embodying torque converters. Other makers who until recently offered only a three-speed synchromesh transmission with possibly a "kick down" overdrive, have omitted the intermediate stage of fluid coupling in conjunction with a "stepped" gearbox, and have introduced torque converter transmissions giving fully automatic control.

A Detailed Survey

Many of these transmissions have been described in some detail, but so far a comprehensive review dealing as much with the underlying design characteristics as with the detail arrangement has not been published. In this issue there appears the first part of an article that is intended, not to describe the detail operation of the various American transmissions at present in production, but to provide a background of fundamental design characteristics against which the differences between the various approaches to the problem will stand out.

Even among American vehicles which are so often considered to have very similar performance characteristics, there are wide enough variations in power/weight ratio to require individual solutions to the problem. Notwithstanding the superficial similarity in that all the systems described embody a torque converter in association with planetary reduction gearing, there are, in fact, substantial fundamental differences, the reasons for which are in some cases obvious, but in others give rise to some justifiable speculation. Thus, on vehicles of high power/weight ratio the whole of the torque multiplication that may be required in the normal driving range is provided by the torque converter, and the planetary reduction is brought into use only in extremely severe conditions. In the case of vehicles having an appreciably lower power/weight ratio, it is found necessary to provide a planetary reduction in the normal driving range, giving in effect a two-speed transmission but also with an emergency low range. Both these main divisions, however, can be again sub-divided into those in which direct drive is provided by the torque converter operating above its coupling point, and others in which the converter is "locked out" at the coupling point by a clutch that provides direct drive.

Not enough reliable data can so far be obtained to make an entirely convincing case for either side, and it would appear that the question can only be answered by further practical experience under widely varying road conditions.

Technical Achievement

While full credit must be given to American manufacturers for the development of these highly complex mechanisms to the point where they can be offered at a figure of $160 to $200 above the basic price of the car, it must be remembered that some sacrifice in overall efficiency can be tolerated in the American conditions of plentiful and relatively cheap fuel supplies. Further, these conditions have always resulted in an average power/weight ratio among American cars appreciably higher than in most other countries.

These basic factors must be kept well to the fore when considering the adoption of similar transmissions on British cars, for example. Some thought might also be permitted on the question of whether these transmissions are the result of an insistent public demand or whether the demand was created or, at any rate fostered, by the sales organisations.

While the case for the torque converter can perhaps be considered convincing in the case of those British cars having power/weight ratios comparable with the largest American cars, the majority fall appreciably below this standard. It is to be expected that with medium and low power/weight ratios the increased fuel consumption resulting from a torque converter transmission operating off the coupling point for appreciable periods of time, would not be acceptable in countries where fuel is expensive. There is some evidence that with a high power/weight ratio the

increased fuel consumption is scarcely noticeable on long main road journeys wtih a lightly loaded car. With a fully laden car, however, an increase in fuel consumption of the order of at least 10 per cent. is probable. In the absence of truly comparative figures for similar cars, the figure of 10 per cent. must be accepted with reserve, and it seems likely that it is a conservative estimate. A higher proportion of traffic driving or operation on second class roads and in hilly districts would doubtless show similar results.

Other problems associated with torque converters concern such matters as engine silence, over-run braking, etc. Much attention has been paid to these points in the U.S.A., and the use of self-adjusting tappets, for example, has become general for these applications, and doubtless problems have been encountered in the matter of intake and exhaust silencing and the general question of rendering the power unit as remote as possible from the passenger space.

Fuel Consumption

These and similar problems, however, are perhaps secondary to the problem of fuel consumption so far as British cars are concerned. At the moment it seems probable that the low efficiency of the torque converter operating at a torque ratio of about 1.5 or below would prove unacceptable. It seems doubtful whether any form of fluid drive other than a straightforward fluid coupling in association with a "stepped" gearbox could maintain the desirable high efficiency over the necessary range of torque multiplication.

So far the broad question of whether fully automatic, semi-automatic or manual control is desirable has been left open. There is little doubt that an expert driver with the ordinary synchromesh transmission can not only obtain the maximum performance of which the vehicle is capable, but can also obtain it with a smoothness equal to that of a torque converter system in which no change of gearbox ratio is made in accelerating from rest. In the case of the reasonably competent driver who does not study refinement of control to such a great extent, there is probably little to choose in smoothness between his handling of the car and the operation of an automatic transmission in which changes of reduction gear ratio are also made. For the great majority of car users, however, the fully automatic transmission undoubtedly has much to offer, and it is probable that they would be prepared to pay an appreciable premium for the manifest advantages.

Although too much significance should not be attached to the following figures, they are of interest as showing the trend on medium-priced vehicles in the U.S.A. During the first ten months of 1950, Buick produced 446,803 cars and 357,918 Dynaflow transmissions. Doubtless a proportion of the Dynaflow figure can be disregarded, since this transmission has been in production since 1948 and presumably a number of service units are included in this total. Nevertheless, it would appear that perhaps 65 to 70 per cent. of Buick customers are prepared to pay for the automatic transmission.

Reverting to the British car there is one point upon which there seems to be a substantial measure of agreement. This is the elimination of the clutch pedal, the correct operation of which remains by far the most difficult problem facing not only new drivers but, it would seem, many who have driven for a substantial total mileage.

Tyres and Design

The coming of the low pressure tyre had considerable repercussions on suspension design. This must obviously be a natural development, since tyres of this type are much more a part of the suspension than those of the old pattern, small diameter, high pressure type. Having become then, essentially the most important basic part of the vehicle suspension system, it is not surprising that the satisfactory functioning of the whole system depends almost wholly upon tyre optimum pressures. Not only must these be just right and maintained so within narrow limits, but the ratios front and rear will vary widely with every vehicle, even amongst those of similar types. In some cases it is found that the designers and the manufacturers themselves do not always appreciate quite how critical these pressures are, even on modern products. A pound or two either way will make all the difference between road holding and wandering, as obviously there is a direct tie up between under and over steer effects and the proportion of roll front and rear.

In one vehicle it was found that in order to secure the best possible compromise, the tyre pressures at the front had to be so low as to be almost to the wall stressing danger line, while those at the rear needed to be excessively hard and so produced a tendency to choppiness or shortness in the rear axle movement. The important point is however that this supersensitivity to tyre pressure is of itself a retrograde tendency from the viewpoint of the ordinary user. Very few people either here or abroad, can be relied upon to maintain tyre pressures sufficiently near to the ideal to secure optimum performance of the vehicle, and maximum life and mileage of the tyres themselves. More control of roll front and rear and consequently under and over steer characteristics should be built in to the mechanical side of the design rather than employ finely calibrated tyre pressures as a means of effecting the necessary adjustment.

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P.N.D.L.R.

A Detailed Study of the American Torque-Converter Transmissions

By O. D. North, M.I.Mech.E.

THE somewhat cryptic initials that appear at the head of these notes, are the lettering that has been standardised by the American automobile industry for the controls of their torque converter transmissions. Given this clue, they will be readily understood to indicate Parking, Neutral, Drive, Low, Reverse.

After a period during which it seemed that the four-speed automatic epicyclic transmission, as exemplified by the General Motors "Hydramatic" gearbox, was to be the ultimate connection between the engine and the back axle, the torque-converter transmission became the accepted type.

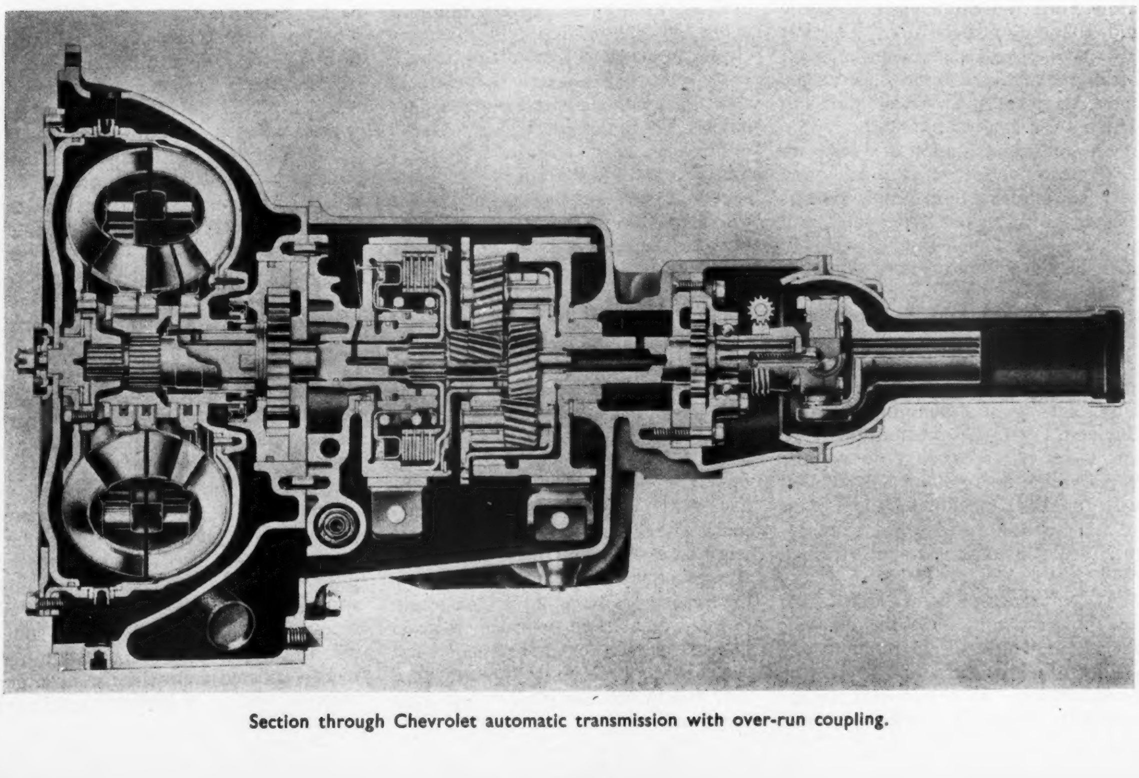
Introduced in its present form by General Motors as an optional extra on de luxe models of the Buick cars, it was then offered, on similar terms, on the Chevrolet range, which is directed to a much lower-priced market. The response was greater than even the manufacturers had expected. Packard came on to the market with a similar mechanism, to be followed by Studebaker and then by Ford on the Mercury range, as a forerunner of its eventual adoption on the other Ford models.

It would give quite an unfair picture of the American automobile industry to suggest that the other makers rushed to copy General Motors, once the public success of the torque converter was assured. So extensive is the development work involved, extending to 4½ years, for example, in the case of Studebaker and so lavish is the special equipment required for the quantity production of an exceedingly complicated mechanism that a sudden and opportunistic "switch-over" was quite out of the question. The amount of enthusiasm and hard work that has gone into the development and tooling of the various forms of the same basic idea merits the profound respect of all automobile engineers. There has been then, a notable change in American automobile practice, one that is no "flash in the pan" and one that must have important reactions on the makers and sellers of automobiles the world over.

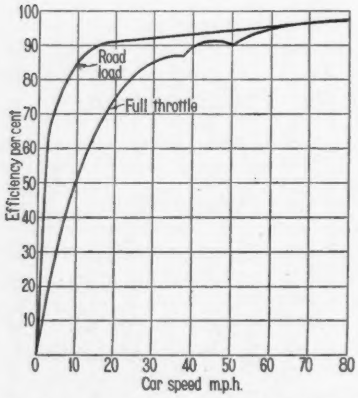
The torque converter itself is, hydraulically, a complicated compromise. American engineers, largely by experiment and trial rather than abstruse mathematical reasoning, have succeeded in making considerable advances in working efficiency. They have moreover shown considerable ingenuity in getting round some of the defects inherent in the principle. They have not been ashamed, in some cases, of producing a mechanism relying as much for its efficiency on a step-by-step epicyclic gearbox as on the torque-converter with which it is associated.

While the basic principle is completely accepted, design details are in a state of flux and further progress is probable. This will be facilitated by the immense scale of American production. In spite of lavish tooling, output is so great that equipment overheads are soon wiped out. The introduction of modifications often means only the replacement, by different jigs and tools, of plant already worn out.

In considering the potentialities of torque - converter transmissions in English conditions, certain qualifications must be made. In America, fuel is relatively cheap and a consumption of, say, 15 miles to an English gallon is not a serious handicap to a vehicle. At present there is a large unsatisfied demand for cars in



Section through Chevrolet automatic transmission with over-run coupling.



Efficiency/car speed curves for full throttle and road load of typical torque converter.

America and plenty of money with which to buy them. An extra charge of about 200 dollars for torque-converter as against the orthodox gearbox made little difference to the insistent demand for the latest transmission.

It must also be remembered that all American cars have a high power-weight ratio and, in many cases, where a torque converter is offered as an alternative, a larger engine is fitted and the compression-ratio is increased as well. None of the systems has less than 100 h.p. at disposal; even the low-priced Chevrolet has 105 gross h.p. provided by its specially enlarged high-compression 3.860 litre engine. Buick do not offer the system with less than 122 h.p., Packard provide 138 h.p., while Ford and Studebaker have 100 and 102 h.p. respectively. The last two have three-speed epicyclic gearboxes capable of relieving the torque converter of the awkward situations, in which its efficiency tends to be low.

The fuel consumption of a car with a torque-converter varies much more with conditions of operation than is the case with an ordinary clutch and gearbox. In fairly fast driving on open roads without many stops or traffic checks, the increase in consumption is slight. It falls off badly where there is much driving in traffic and under the checks and restraints involved in driving on some of the hilly, narrow and winding roads of England. In these conditions, the torque-converter car may do as little as 14 miles per gallon as compared with 18 miles per gallon for an identical vehicle with a three-speed synchromesh gearbox and standard friction clutch.

Also it cannot be denied that with a low power-weight ratio the case for the torque-converter is much weakened if fuel consumption is a vital consideration. In a relatively under-powered vehicle a torque-converter tends to show up as a second-rate fluid coupling combined with a third rate gearbox as far as efficiency is concerned.

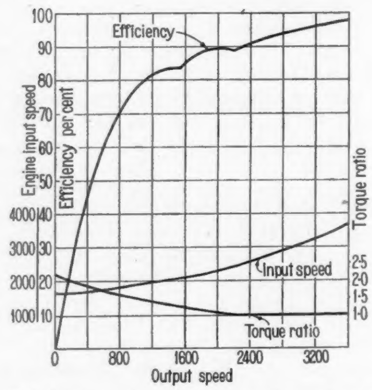
This is very well brought out in the opposite case of the high-powered American vehicle, for which two curves of efficiency plotted against car speed are given, one for full throttle and the other for what the Americans so aptly term "road load", in other words, the power required to overcome rolling and windage resistance only at any particular speed.

Converter Efficiency

In the example given the car has what is known as a "polyphase" converter, in which two stators, with vanes at different angles, are provided. Both of these are, of course, mounted on free-wheels and, as the curves indicate, one goes out of action at about 38 m.p.h. at full throttle and the other at 50 m.p.h. from which point onwards the converter functions as a fluid coupling with about 7% slip at 50 m.p.h., which falls to some 3% at 80 m.p.h. At 30 m.p.h. corresponding to 1,350 r.p.m. of the output shaft, the efficiency is about 83%, with the converter in operation. It gives a torque multiplication of about 1.35 to 1 with an engine speed of about 2,000 r.p.m. and an engine horsepower of about 85.

At 20 m.p.h. corresponding to 900 r.p.m. of the output shaft, the efficiency is only 75%, with a torque multiplication of 1.55 to 1 and an engine speed of about 1,850 r.p.m. corresponding to about 78 engine h.p. It must be clearly understood that on full throttle every multiplication ratio is inevitably associated with a particular engine and output shaft speed. Thus, for a multiplication of 1.55 to 1, roughly that of many third speeds in four-speed English gearboxes, not only is the maximum efficiency set at 75%, but the engine is held down to a speed at which it can only develop about two-thirds of its maximum horsepower.

Some 59 horsepower are delivered to the back axle, whereas with an ordinary gearbox, efficiency say 95%, and a gradient exactly fitting the gear ratio, the engine might be running at 3,600 r.p.m. developing 122 h.p. and delivering no less than 116 of it to the back axle. While the comparison may seem rather unfair, it will serve to emphasise the point that the practical success of the torque-converter lies in the fact that, owing to the generous power-weight ratio provided, it never has to operate for more than a few seconds at a time under the unfavourable conditions assumed in our comparison.



Full load performance curves of typical torque converter.

On full throttle at 20 m.p.h., the actual car under consideration would have a tractive effort giving it an acceleration of about 5 feet per sec. per second, or capable of driving it up a 15% gradient. In about three seconds of full throttle on a level road it would be doing more than 30 miles an hour. The acceleration provided is about as much as is comfortable for passengers not bracing themselves for high-speed motoring. Moreover it is continuous, with no interruption of engine power and acceleration for a change into a higher gear.

Largely on account of this continuity of torque, the actual acceleration from rest to maximum speed does not take appreciably longer than with an orthodox gearbox expertly handled. Moreover, on such a car assumed to be fitted with a three-speed gearbox, it is impossible, under most conditions, to use anything like full engine power on first speed without giving rise to wheelspin and a most unpleasant sensation for the occupants.

In American automobile design circles, the standard of comparison for torque-converter transmissions is with a start on second speed of a three-speed gearbox. In this case, if the clutch is gently engaged to avoid jerk, the actual efficiencies, allowing for clutch slip, are not so much better than the torque converter can provide.

In the example under consideration the "stalled" torque multiplication is 2.25 to 1, holding the engine down to about 1,500 revolutions per minute. One of the features of the torque-converter is that it is impossible, in normal driving, to load the engine down to a very low speed. A higher compression-ratio can therefore be safely employed without risk of pinking and this point is exploited by all manufacturers. It offsets, to some extent, the increased fuel consumption to be expected from the use of larger engines and a relatively inefficient transmission.

That the comparative inefficiency, which ultra-scientific engineers cannot overlook, has in practice less importance than might be imagined, can also be observed from the "road load" curve. From this it will be seen that at steady speeds on a level road the throttle requirement and corresponding engine torque is such a small proportion of the maximum at low and moderate speeds that the converter hardly comes into action at all. From 20 to 80 miles per hour the transmission is operating as a fluid coupling, with rather a bad slip at 20 m.p.h. There is no actual torque multiplication, both stators freewheeling all the time. The efficiency remains above 90% and some of the loss which this represents is accounted for by the power taken to drive the two fairly large oil pumps which provide the operating pressures for the epicyclic gearbox.

Transmission Systems

In three of the systems, Buick, Chevrolet and Packard, the torque-converter is the sole multiplier used in normal driving. All systems, however, employ a low ratio in the form of an epicyclic gear which can be brought into action, under full throttle if desired, by a lever on the steering column. A ratio of 1.82 to 1 is given on the Buick, Chevrolet and Packard, the Ford-Mercury and Studebaker giving 2.44 and 2.31 respectively.

Speeds up to about 40 m.p.h. can safely be attained in what is termed the "Low" position and it is intended for use in a very hilly district, particularly if the winding nature of the road involves checking car speed and accelerating again on a steep gradient. The converter is then relieved of torque conversion to such an extent that it runs as a coupling practically all the time and the efficiency is quite high.

The "Low" position can also be used in accelerating from rest if a very quick start is desired, the driver running up to about 30 m.p.h. in "Low" and changing up to "Drive" position without releasing the accelerator. All the gearboxes have special provision in one form or another for engaging direct drive while low gear is still taking the load. This also eliminates engine racing on the change over and at the same time minimises any "lock-up" effect due to two gears being engaged at once. At no time, therefore, during acceleration from rest on "Low" and up to maximum speed on "Drive" need there be any break in the power flow.

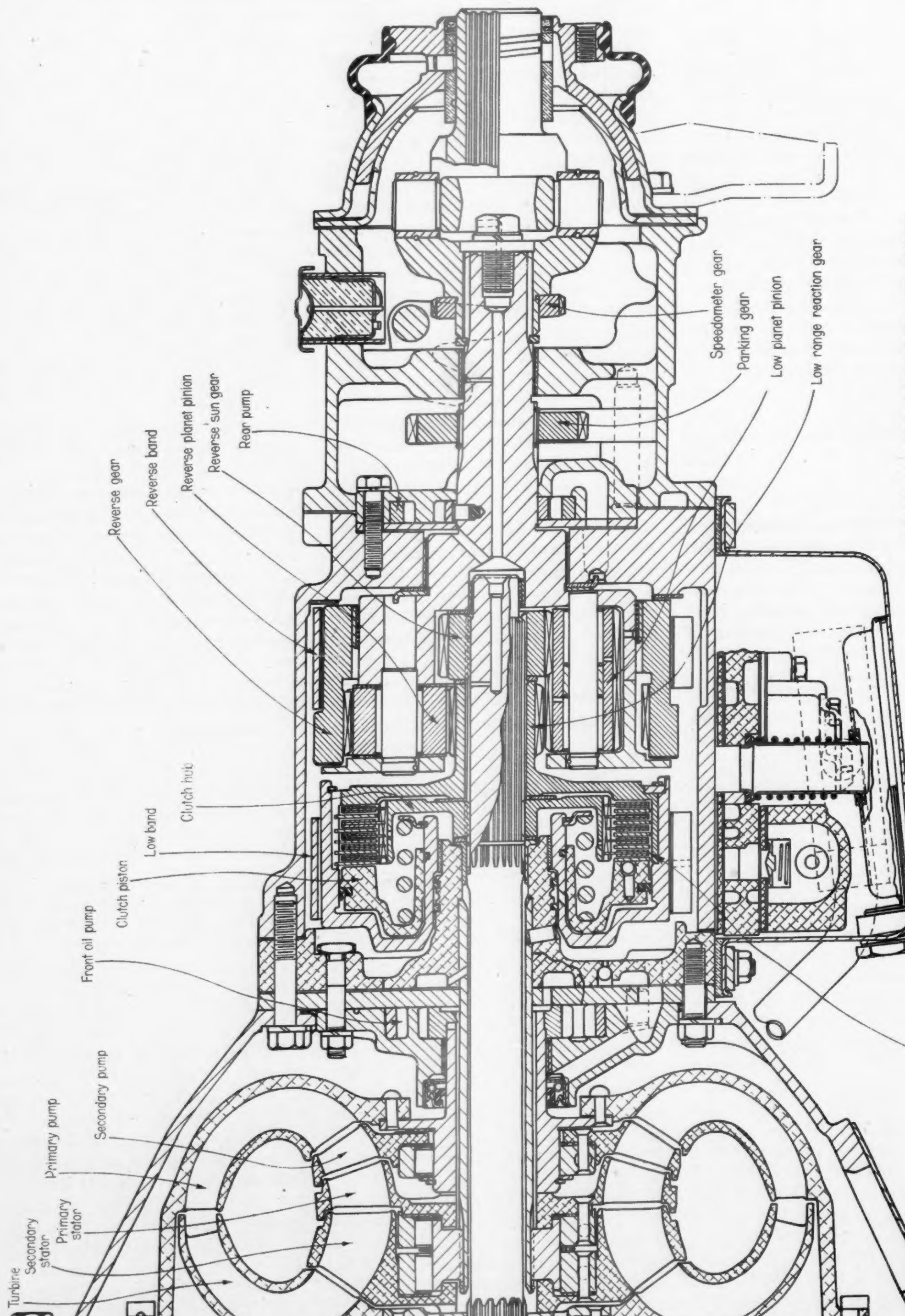
In both the Ford-Mercury and Studebaker systems three-speed gearboxes are employed and the intermediate gear is automatically engaged and disengaged by governor mechanism based on a combination of car speed and throttle position. A ratio of 1.48 and 1.435 is employed, and, broadly speaking, the coupling is never called on to work at a torque multiplication of more than about 1.4 to 1 in the "Drive" position. The mechanism changes down into intermediate gear at this point.

The converters can be somewhat simpler, since a high "stall" torque is not required and efficiency at the greater torques is not so important. Somewhat better acceleration is given in "Drive" position, and also better hill-climbing. This is important, as the power-weight ratios of both Ford and Studebaker are not as high as for the other three, the engines giving 100 and 102 h.p. respectively.

	Buick Special	Chevrolet	Packard 200 de Luxe	Studebaker	Ford-Mercury
Converter type ..	Polyphase. 2 Stators, 2 Pumps	Polyphase. 2 Stators, 2 Pumps	Simple Pump. Simple Stator Double Turbine	Simple.	Simple.
Torque ratio, stalled ..	2.25 to 1	2.25 to 1	2.4 to 1	2.15 to 1	2.1 to 1
"Drive" range bracket	Converter only	Converter only	Converter locked or converter	Converter locked or converter x by 1.435 intermediate gear	Converter or converter x by 1.48 intermediate gear
Axle ratio	3.9 to 1 (4.1 to 1)	3.55 to 1 (4.11 to 1)	3.9	3.54 to 1	3.31 to 1 (3.90 to 1)
Tyres	7.60 × 15	6.70 × 15	7.60 × 15	7.60 × 15	7.10 × 15
Kerb weight	4,020 lb.	3,400 lb.	4070 lb.	3,500 lb.	3,620 lb.
Engine.					
Capacity—cu. ins. ..	248	235.5	288	245.6	255
litres ..	4.067	3.860	4.720	4.030	4.185
Maximum torque ..	216 lb. ft at 2,000	193 lb. ft. at 2,200	—	205 lb. ft. at 1,200	—
Maximum B.H.P. ..	122 B.H.P. at 3,600	105 B.H.P. at 3,600	138 B.H.P. at 3,600	102 B.H.P. at 3,200	100 B.H.P. at 3,600
Compression ratio ..	(P) 7.2 to 1 (6.6 to 1)	6.7 to 1 (6.6)	7.5 to 1	7.0 to 1	6.8 to 1
Planetary reduction in "Low" range	1.82 to 1	1.82 to 1	1.82 to 1	2.30 to 1	1.48 to 1 and 2.44 to 1
Special braking provisions	None	Overrun fluid coupling	Converter lock on overrun on 'Drive' and 'Low' ranges	Converter lock on overrun on 'Drive' Specially low bottom gear	Specially low bottom gear
Converter cooling ..	Water-jacketed cooler on transmission	Oil cooler in radiator outlet	Oil cooler in radiator bottom tank	Direct air-cooling of converter body	Direct air-cooling of converter body

Notes :—Figures in brackets are for corresponding car when fitted with normal 3-speed gearbox.

(P) Signifies Premium Spirit advised on this engine.

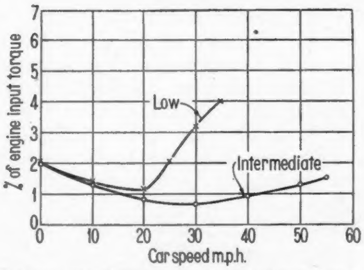


GENERAL ARRANGEMENT OF BUICK AUTOMATIC TRANSMISSION.

As may perhaps be evident, one of the weak spots of the torque converter is the change-point from torque multiplication to fluid coupling. The simple converter, with only one stator, evidences a drop to about 85% efficiency at a 50 to 60 m.p.h. on full throttle. This condition may obtain for considerable periods in a moderately hilly district.

On the Buick and the Chevrolet the problem is tackled from the converter end, cutting out much of the "droop" in the curve by using two stators with different vane angles, making the change-over less sudden. Packard and Studebaker however adopt the radical policy of locking the torque converter solid above speeds determined by governor mechanism interconnected with the throttle. At small throttle openings the direct-drive clutch remains engaged down to about 15 m.p.h. while when the accelerator is released the clutch remains engaged down to about 12 m.p.h. Full-throttle at moderate speeds disengages the clutch, allowing the torque converter to take over and incidentally reducing the liability to "pinking" caused by hard pulling at low speeds. At any speed below 50 m.p.h. full depression of the accelerator pedal releases the clutch. Normally this is automatically engaged and remains so above 55 m.p.h., since the converter offers no advantage after this speed is attained.

On the Studebaker, the engagement of the direct-drive clutch is under similar control, but at its release the



Torque requirements of spinning direct drive clutch 9½in. diameter.

torque-converter is driving through the epicyclic reduction of 1.435 to 1. There is therefore a definite step-change, the operation being made shockless by the provision of a free-wheel in the gearbox.

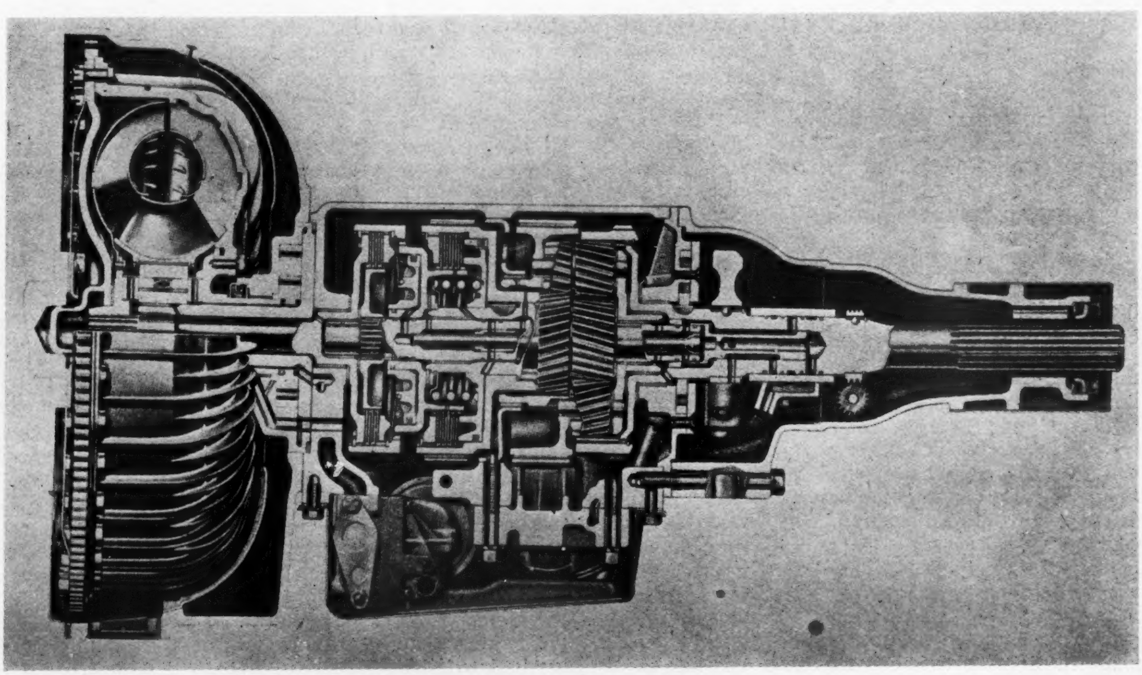
The friction clutch, as in the case of the Packard, runs immersed in the oil of the converter; it is hydraulically operated and is 9½" in diameter. While it cuts out slip losses when engaged, it introduces drag losses when free, owing to the oil movement, as shown in the curve, which is from H. E. Churchill's paper before the S.A.E. It may be that the author of this paper does the device less than justice in this curve. If the figures represent the drag torque on the clutch shaft one part in 1.435 should be credited back on intermediate gear. The clutch drag, unamplified by the gearbox or converter, does add unit value to the output torque, so that the losses may not be so great as shown. Among the precautions taken to reduce drag are a big running clearance and a definite restriction of the flow from the converter circulating feed into the clutch chamber.

The effect of the step-change in normal driving is to reduce very much the duty imposed on the torque-converter, which, with the 1.435 reduction can handle most loads while functioning as a fluid coupling and will very seldom be called on to work on the inefficient parts of the curve.

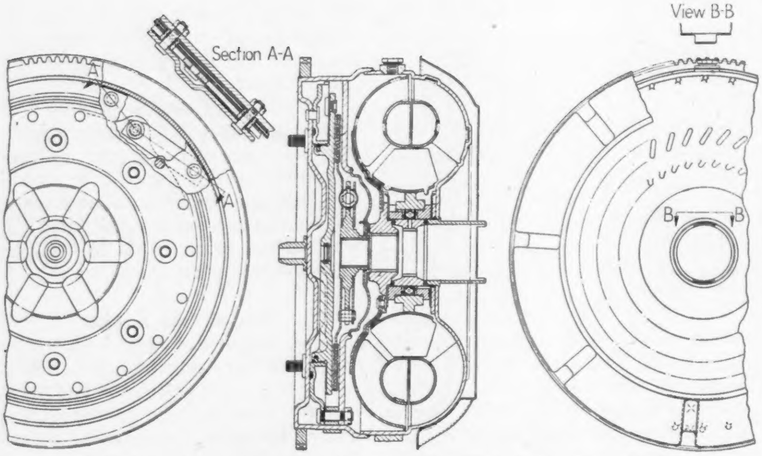
Engine as Brake

Since intermediate gear is picked up through a free-wheel the reduction it provides is not available on the over-run for braking. On release of the accelerator pedal, however, the direct-drive clutch engages so that the engine is available as a brake. This availability of the engine as a brake has engaged the serious attention of American engineers. It will be understood that the torque-converter is not double-acting. Since the stators are mounted on free-wheels there is no torque multiplication on the overrun. Moreover, the converter is worse than a modern fluid flywheel in transmitting power from the output shaft to the engine flywheel and the slip is greater than when driving. There is also a definite lag in the build-up of braking torque, the impression given to the driver being described by some American engineers as "cushioned backlash".

No torque-converter transmission gives as effective engine braking as first speed of the standard three-speed



Section through Ford-Mercury automatic transmission.



Studebaker three element torque converter with direct drive clutch. (Long Manufacturing Div., Borg-Warner Corp.).

gearbox. Some of them, in the early stages of development, were very much worse than this, reaching, for example, terminal coasting speeds of about 30 m.p.h. on a 1 in 10 gradient with "Low" ratio engaged. There appear to be three different approaches to the problem. Ford-Mercury and Studebaker fit epicyclic gearboxes in which the "Low" train gives a greater reduction than that required from the hill-climbing standpoint, these being 2.4 and 2.3 to 1 respectively. There is little objection to what may be regarded as unnecessarily low first speed in the case of these two cars, since both of them have three-speed gearboxes, the intermediate gears of which provide overall reductions, when added to the torque-converter stall figure, of 3.11 and 3.08 respectively. "Low" ratio is therefore very rarely required, though it may be mentioned that "house trailers" weighing some 4,600 lb. appear to be in common use in America. One of the standard tests of one American maker is equivalent to towing such a trailer up six miles of a 6% gradient.

Packard get over the difficulty by arranging that their direct-drive automatically-operated clutch comes into action on the overrun on both "Drive" and "Low" ratios above 12 m.p.h., thus eliminating converter slip and keeping the engine speed high enough to dissipate the energy. Buick make no special provision, but on the Chevrolet, introduced later than the Buick, the torus of the converter is fitted with a special overrun fluid coupling. This consists of two sets of radial vanes assembled in opposed channel-section pressings, one set being attached to the turbine, or output member, and the other to the pump member which is bolted to the engine flywheel. The vanes in the turbine are arranged at a slightly larger radius than those in the pump.

When driving, the pump always runs faster than the turbine, but since the vanes attached to it are at a smaller radius they are unable to initiate any appreciable vortex. On the overrun, however, the pump runs slower than the turbine and the vanes in the turbine, being at a larger radius, set up a rapid circulation and give considerable coupling effect. There is, of course, some drag when driving, but, as mentioned in connection with the Studebaker locking clutch, this is not all loss, appearing, in unmultiplied amount, at the output shaft.

The importance attached by American designers to adequate coupling between back wheels and engine on the overrun is partly to assist what is known as the "taxi-push start." It is apparently common practice for owners in difficulty in the winter to ring up the local taxi garage to give them a push. For this, among other reasons, all torque-converter cars have two oil pumps supplying the control servos, the first one driven by the engine and the second one by the output shaft, thus ensuring oil pressure when the engine is at rest.

The design of the torque-converter and its associated gearbox and controls is still in a state of flux and much constructive argument is to be found in the American technical press to the relative merits of different systems. For example, the Studebaker engineers claim that their locking clutch economises in fuel by cutting out fluid coupling slip and the earlier stages of torque conversion in the region of the change-point. As against this, the designers of the Ford-Mercury claim that by omitting the locking clutch they can use a higher back axle ratio without the "pinking" at moderate speeds and loss of acceleration that would result if a high ratio were held by a locking clutch.

Ford-Mercury use an axle ratio of 3.31 to 1, with the converter always in circuit and an intermediate ratio of 1.48 automatically available in "Drive" range. Studebaker lock out their torque-converter at high speeds and small throttle openings and fit a back axle with a ratio of 3.54 to 1, the torque-converter driving through an intermediate reduction of 1.435 to 1 when the locking clutch is released.

Axle ratios in general are on the high side where torque-converters are fitted. Ford-Mercury, for example, fit an axle of 3.90 to 1 ratio if an orthodox three-speed synchromesh gearbox is used. All the cars are capable of about 90 m.p.h., top speed being attained slightly beyond the peak of the engine power curve. The ratios employed are not as high as those given by the semi-automatic overdrives still available as alternative equipment.

One of the features of an unlocked torque-converter is that engine speed changes slightly but quickly with every movement of the accelerator, dropping perhaps 15% between full throttle and idling, even when the car speed is such that no appreciable torque-conversion is taking place. Being no longer "geared to the back wheels", the engine note and other noises become more obtrusive. Realising this, makers in general are paying special attention to sound insulation on torque-converter models, and hydraulic tappet adjusters are generally included in the specification, though they may not be fitted to the standard models with normal gearboxes.

Manual Control

Control for selecting "Low" or "Drive" range, for obtaining reverse, neutral and parking positions is always by a small steering-column lever. This operates a multiple-landed piston valve controlling the oil supply to the various clutches and brakes. With the exception of Ford, all makers have standardised the indications of control lever position, the legend reading P.N.D.L.R. In the P, or Parking, position the gearbox is in neutral, the engine starter relay circuit is "live" and, in addition, a spring-loaded pawl is engaged with a toothed wheel on the output shaft, locking the vehicle at rest. This provision may seem to be a reflection on the handbrake which is also fitted for the same purpose. The makers point out, however, that it is much simpler to operate one lever instead of two and that it has not been unknown for drivers to omit to release the hand-brake, with disastrous results to the rear brake linings and hydraulic

cylinders. The locking pawl is generally so rounded that it will not stay in engagement until the vehicle speed has been reduced to one or two miles an hour. Studebaker provide a hydraulic pawl-locking cylinder connected to the rear oil pump which makes it impossible to attempt the pawl engagement until the vehicle is actually at rest and the oil pressure has dropped to zero.

The "N" position also leaves the starter relay circuit "live" and the engine can thus be started in either P or N position, but not in any other. The locking pawl is disengaged and the car can be towed or pushed.

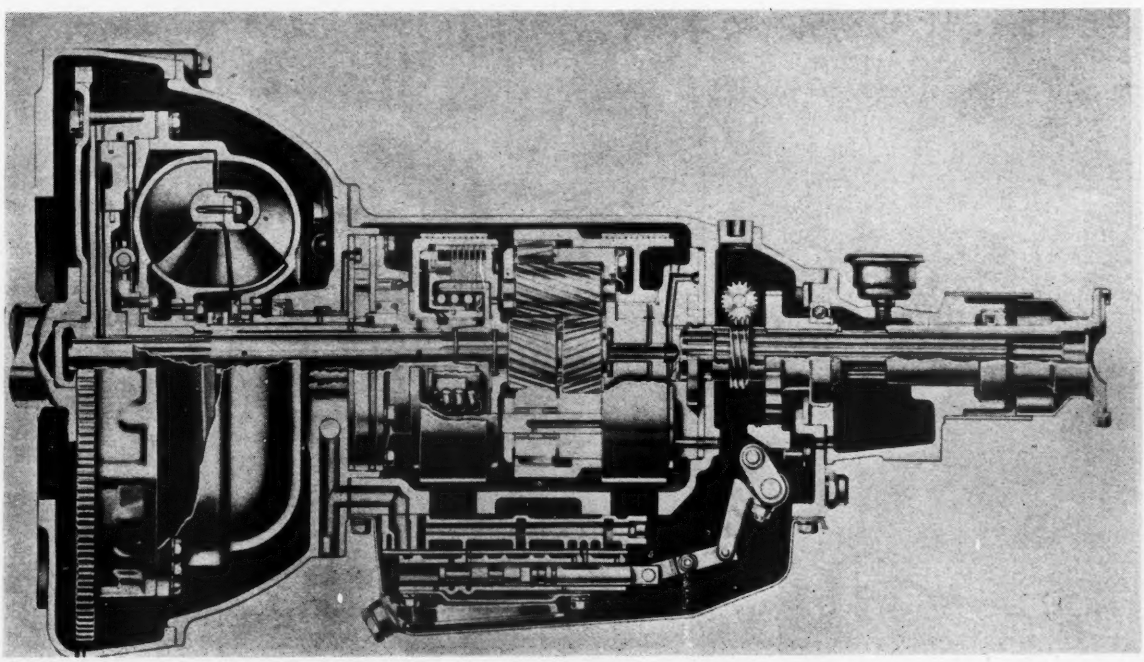
In the D or "Drive" position, the gearbox is locked in direct drive, the reduction given by the torque-converter alone being available for acceleration and hill-climbing. In Ford and Studebaker the band of the intermediate gear is held when the lever is moved to D position and the car is at rest. Starts are made on intermediate gear with the benefit of converter torque multiplication, further multiplied by the epicyclic reduction. Under control of car speed and throttle position the change into direct in the gearbox is made automatically as the car gains speed. Ford as previously mentioned, retain the use of the torque-converter after the epicyclic is locked solid. Studebaker, on the other hand, clutch right through from flywheel to output shaft, cutting out the torque-converter and the epicyclic reduction in one operation. Packard start on the converter alone and lock it out at a suitable speed under governor and throttle control. The L position gives the lowest reduction available in the epicyclic and the converter torque multiplication as well. In the case of Ford only, the governor changes into intermediate gear above 40 miles per hour, regardless of throttle position. Intermediate gear on the Ford is thus available both as the lower end of the "Drive" range and as the upper end of the "Low" range, very much extending this and rendering it specially suitable for use in hilly country with varying gradients.

The last, or R position, engages the reverse gear band. On all the cars except Ford, the reverse position comes next to the low, with no neutral between. This is intended to assist in "rocking" the car between low and reverse when extricating it from a hole in mud or snow. Ford however, place the reverse between parking and neutral position, their notation being P.R.N.D.L. It is claimed that this arrangement eliminates the risk of accidental engagement of reverse when moving from "Drive" to "Low" positions. As an additional safeguard it is necessary to lift the lever over a stop to get out of neutral or "Drive" positions. Studebaker embody an ingenious safety lock to prevent accidental engagement of reverse while the car is moving forward at more than about 5 m.p.h. The delivery from the rear pump, which is driven from the output shaft, is connected to a spring-loaded piston valve which vents the reverse servo oil line as long as the rear pump is delivering appreciable pressure.

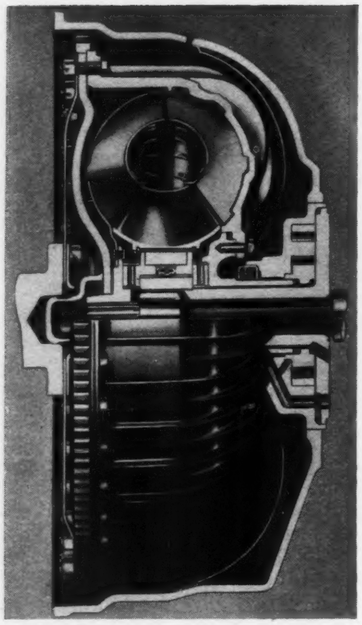
As the car slows down the pressure drops, allowing the piston valve to close the port and permitting the reverse-servo to operate normally. This arrangement does not prevent the car from being "rocked" without any momentary pause in neutral, the piston valve in fact "takes charge" of the operation.

While on the subject of controls it should be remembered that there is no clutch pedal, or equivalent, on the torque-converter car. The accelerator pedal is always operated by the right foot while the brake pedal can be worked by either foot as desired. Studebaker make deliberate provision for this, the pedal plate being much wider than normal so that the left-hand part of it comes conveniently to the left foot.

It is not yet customary to drive with the left foot over the brake pedal, and complaint is frequently made of "traffic-light" creep. For such frequently-repeated stops the control is always left in "D" position. A fast tick-over, such as is automatically provided by some carburetters when cold, will however transmit appreciable power through the torque-converter, so that the car creeps forward. Studebaker make elaborate provision against this by a solenoid-operated check valve in the rear brake pipe line. This solenoid is wired in series with two switches, one of which is closed when the accelerator-pedal is released, while the other closes by drop in rear



Longitudinal section of Packard automatic transmission with direct drive clutch.



Ford-Mercury three element torque converter.

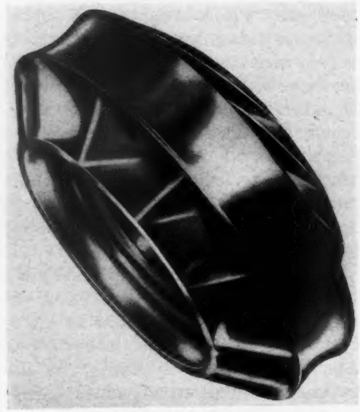
pump pressure as the car comes to rest. After the vehicle has been brought to rest, therefore, the rear brakes are held on, preventing "creep", until the accelerator pedal is slightly depressed for a restart.

Converter Cooling

Regarding the cooling of the torque-converter, while at fairly high speeds, with the converter acting as a fluid coupling and moderate torque giving reduced slip, the losses are not seriously above those of an ordinary gearbox, under severe conditions there is a considerable amount of heat to be dissipated. Buick, Chevrolet and Packard circulate the converter oil through coolers jacketed by the engine cooling water. On the Chevrolet, the radiator has a 10% increase in cooling surface by the closer spacing of the air cells. Its heat dissipation capacity is still further increased above that of the standard gearbox car by the fitting of a radiator pressure cap set to 4 lb. per square inch.

The oil cooler is located at the radiator water outlet connection. It consists of two opposed pressings forming a water jacket within which is a flat oil chamber. This chamber is reinforced against bursting pressures on its flat walls by a zig-zag perforated pressing. This is copper brazed to the inner walls of the jacket in such a manner as to give multiple cross-ties. It provides additional surface for heat transmission from the oil to the metal, as compared with the metal to water surface. A thermostat in the oil circulation system by-passes the cooler until the oil temperature rises to 240 deg. F., thus maintaining a high oil temperature with correspondingly low viscosity so important for torque-converter hydraulic efficiency. On other designs, the oil cooler is on the transmission, the water being piped to it. The Chevrolet arrangement is, however, to be preferred, giving, as it does, a small amount of hot oil surrounded by a large volume of water. Should the car be brought to rest after a period of severe work there is less chance of the hot oil causing local boiling of the water, with a possible steam-lock in the water pump. Packard follow the same principle and place the oil cooler within the radiator bottom tank, which is of large capacity.

Both Studebaker and Ford use Borg-Warner converters in which the bodies are externally ribbed to give cooling surface and act as air impellers. Cooling air is drawn into the centre of the bell-housing and thrown outwards. About 250 cubic feet of air per minute are passed at maximum speed, at which about $\frac{3}{4}$ h.p. is absorbed by the fan blades. The maximum temperature under severe duty is about 220 deg. F. above ambient, or about 300 deg. F. thermometer reading on a warm day. Much of the foregoing information is summarised in the accompanying table. Space will not permit of complete



Ford-Mercury stator member.

descriptions of these intricate mechanisms. For this, each would demand a small volume. All must perforce be removed from the car complete with the attached gearbox for service. Embodying as they do several concentric shafts, a large oil seal and a geared oil pump, it is clear that deflections of the engine crankshaft must not be allowed to disturb alignment. All five converters, which form the flywheel masses of the engines, are therefore spigoted in the crankshafts by a relatively small-diameter push fit. They are driven by large, thin and flexible steel discs, bolted to the crankshaft flanges at the centre, and to the converter body at their outer edges. The outer ring of nuts are removable with the power unit in position in the car after taking off the front lower bell-housing dust shield.

On the Studebaker the starter ring is shrunk on to the stout pressed steel body of the converter, on the other four cars the ring is welded to the driving disc. A feature peculiar to Studebaker only, is that the complete converter assembly, including the top gear locking clutch, turbine and stator with its free-wheel clutch, are sealed by welding at the works. To obtain access, the weld must be turned off.

(To be continued).

BRITISH PLASTICS EXHIBITION AND CONVENTION

THE British Plastics Exhibition is organised by our associated journal *British Plastics*, Dorset House, Stamford Street, London, S.E.1, published by Associated Iliffe Press. The Exhibition will be held at the National Hall, Olympia, from June 6th-16th, 1951, when all the leading firms of Britain's plastics industry will be displaying their products. Concurrently with the Exhibition the Convention will be held. "Plastics in the transport industry" (including papers on the use of plastics in automobiles, aircraft and shipping) is the title of one of the sessions to be included.

A special feature of the Exhibition will be "Plastics—their design and use", covering 2,000 sq. ft. on the first floor. This exhibit, designed and constructed by the Council of Industrial Design, the British Plastics Federation and the organisers, will supply answers to many basic questions on plastics. It will emphasize their outstanding properties and applications and therefore increase the general understanding of the importance of these materials.

ENGINE DETAIL MACHINING

Recent Developments at the Works of Vauxhall Motors Ltd.

INCREASED efficiency, giving increased production at lower unit cost and without additional strain on the workers, is the aim of the re-organisation programme undertaken by Vauxhall Motors, Ltd., Luton. Practically the whole of the plant and the processing methods are in greater or lesser degree involved, but completion will not be reached for some considerable time. The greatest single development is, however, sufficiently far advanced to permit of a description of the production techniques that will be employed. This is the new building that houses the production functions for heavy vehicles and which will eventually also house the passenger engine and gearbox detail machining and assembly sections.

Many factors must be considered in re-organising for greater efficiency, and two of the most important are the methods of materials handling and the actual machining methods. The materials handling methods adopted for the production of Bedford vehicles were discussed in some detail in *The Automobile Engineer* for *January 1951*. The following notes deal with some of the machining techniques.

In the lay-out of the various lines and sections for machining engine details for Bedford vehicles, the overriding considerations have been: (a) that as far as possible the machining of any detail should be completed at a point adjacent to the appropriate assembly or sub-assembly station to reduce materials handling to a minimum, and (b) the number of loadings and unloadings necessary to complete the machining should be as few as possible. At the outset it may be said that (a) has been obtained in great measure. As for (b), the best examples are found in the transfer machines used in the cylinder head lines, but since one of these lines was described fully in *The Automobile Engineer* for *May 1950*, they are not discussed in these notes.

Although transfer machines are not used on any engine components other than cylinder heads, the machining methods adopted for other components do show developments for carrying out a number of operations at one station. These developments are the result of close co-operation between the engineers of Vauxhall Motors, Ltd., and machine tool manufacturers. Of the machine tool manufacturers who have contributed to this large project, particular mention may be made of James Archdale and Co., Ltd.

It is neither practical nor necessary to discuss the whole of the engine detail machining department to illustrate the methods employed. Therefore these notes deal only with the cylinder block machining line for the new large Bedford engine, with a briefer description of the sections for crankshaft bearing caps and cylinder liners, and the connecting rod. This selection has been made because it includes the heaviest and bulkiest component, the cylinder block, which raises problems of handling as well as of machining, and relatively small light components in which the problems are concerned solely with the actual machining.

Cylinder Block Line

A general principle that has been observed in the lay-out of the cylinder block line is that wherever possible open end work-holding fixtures shall be used and be so placed as to be in line with the roller conveyor between the machines. This type of fixture eliminates transverse movement of the work in to and out of the fixture. Instead, the block passes direct from the conveyor into the fixture at one end, and after the operation is completed it is pushed out at the other end on to the next length of conveyor.

For various easily appreciated reasons, there are a good number of operations for which open end fixtures are not feasible. In other cases, the size of the machine may make it impossible to have the machine table in such a position that the work-holding fixture is longitudinally aligned with the roller conveyor. Nevertheless the number of operations at which only longitudinal movement of the block is necessary is remarkable.

Another noteworthy feature of this line is that despite the weight of the cylinder block, it is seldom necessary to use lifting tackle. In fact, lifting tackle is used only at a few stations where for good and proper reasons the machines are so placed that direct feeding from the conveyor to the machine is not practicable. Where it is necessary to turn the block over or to turn it end for end provision is made for doing so without removing it from the conveyor. In addition, the work-holding fixtures generally incorporate eccentrically mounted

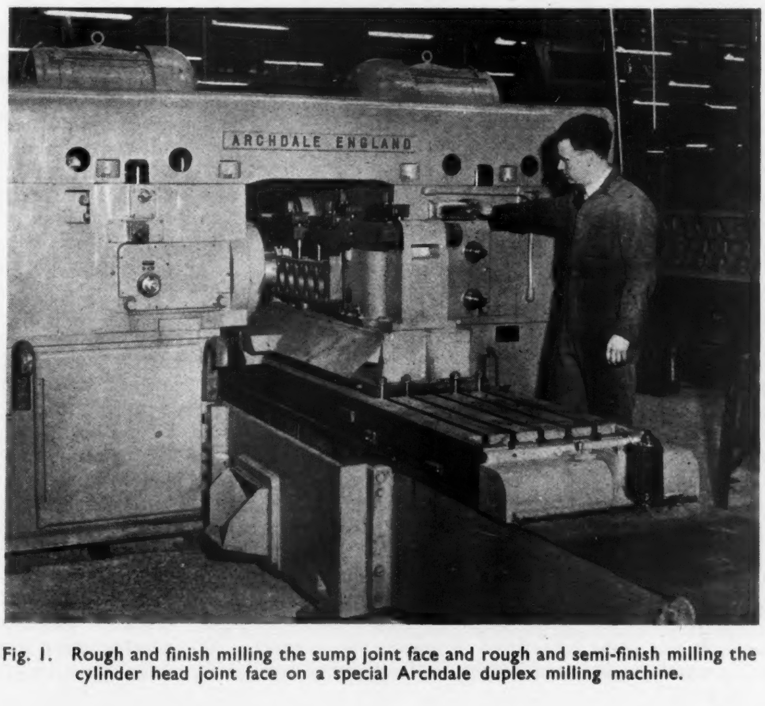


Fig. 1. Rough and finish milling the sump joint face and rough and semi-finish milling the cylinder head joint face on a special Archdale duplex milling machine.

rollers for the purpose of allowing the block to ride easily in to and out of the fixture.

Where it is necessary to turn the block over between operations, a turn-over device is incorporated in the conveyor between the two machines. For example, if at one operation the sump face is presented to the tools and for the next operation the joint face must be brought to the tools, the turn-over can be easily effected without using lifting tackle or removing the block from the conveyor. In such a case the turn-over device is in effect a short length of independent roller conveyor in the conveyor between the machines. It comprises two short lengths of roller conveyor mounted in two steel rings and at a distance apart slightly greater than the height of the block. Each ring rests on two rollers that are carried on ball bearings mounted on a shaft. The block is run into position on to the short length of conveyor in alignment with the general run of the conveyor, and then despite the weight of the component, the turn-over device and with it the block, is easily rotated through 180 deg. to bring the other face into the working position. Where it is necessary to turn the block end-for-end between successive operations, a short length of independent conveyor, that is arranged to rotate about the verticle axis, is interposed between the machines.

Except for two milling operations and a broaching operation and some minor operations with portable tools, all the machining on this cylinder block is carried out on Archdale machines. Most of the Archdale machines are custom built and are special only in that they use combinations of standard units to allow several operations to be performed at one setting. A few are made up from standard units and special units. Three points should be stressed in connection with these machines: (1) the manner in which several operations are carried out at one station by combining heads with different functions; (2) the high rate of metal removal at several operations; and (3) in every case there is a system of interlocks between the locating dowels, the clamps and the machine drive. They are so arranged that until the dowels have given correct location and the clamps are properly applied the machine is inoperative.

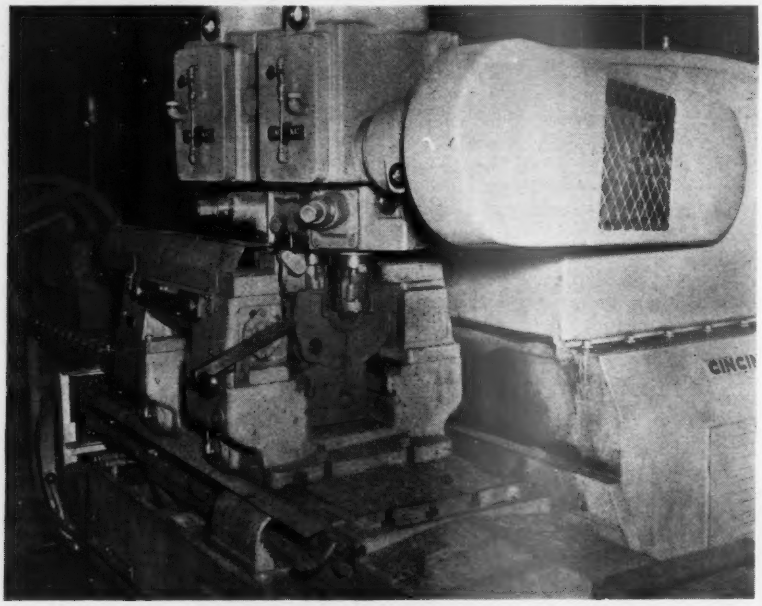


Fig. 2. Set up for milling the bearing cap seats, tenons and side faces on a special Cincinnati machine.

The first of the machining operations is carried out on an Archdale duplex milling machine with five spindles. This machine is illustrated in Fig. 1. It has been specially developed to give a high rate of metal removal and a good quality of surface finish. In one pass through the machine the sump joint face is rough and finish machined and the cylinder head joint face is rough and semi-finish machined. In point of fact, because of the power and rigidity of the machine and the rigidity of the work-holding arrangements, it would be possible to obtain the degree of accuracy and the quality of finish necessary to complete the machining of the cylinder head joint face. There is, however, a danger that this face might incur damage during its passage along the line. Therefore, it is considered advisable to leave 0.030in. to be removed at one of the last operations in the machining sequence.

Three cutters are mounted on one head for machining the sump joint face. They comprise two 8½in. diameter Carbishear cutters with 14 teeth for roughing and a 15in. diameter Carbifine cutter with 36 teeth for finishing. The roughing cutters work at a cutting speed of 252ft./min. and a tooth load of 0.136in. with a 0.125in. depth of cut. To give maximum rigidity, the 15in. diameter finishing cutter has a face plate spindle mounting. It is used at a cutting speed of 354ft./min. and a tooth load of 0.008in. On the other head there is an 11⅝in. diameter Carbishear cutter with 20 teeth for the roughing cut and a 10in. diameter Carbifine cutter with 20 teeth for the semi-finishing cut on the cylinder head joint face. Roughing is carried out at a cutting speed of 262ft./min. with a tooth load of 0.013in. and finishing at 358ft./min. with a tooth load of 0.008in.

High power and high feed rates are outstanding features of this machine. Each head is driven by its own 40 h.p. motor. With the feed and other motors, the total horse-power is approximately 100. A feed range from 20in. to 60in. per minute is provided. In the construction of this machine a standard bed unit and table and, for the smaller cutters, standard spindles and housings are used.

Because of the heavy cut and the high cutting speed, the work must be held very rigidly, otherwise the necessary dimensional accuracy and quality of finish could not be maintained, and in addition, tool life would be adversely affected. Initial clamping is effected by two specially heavy air-operated clamps. It is reinforced by additional manual clamping.

A considerable quantity of swarf is produced at this operation and special arrangements have been made to prevent any accumulation on the machine table or the work-holding fixture. As can be seen from Fig. 1, a sheet metal guard at a steep angle is attached to the side of the work fixture base. As the swarf falls from the cutter it runs down the guard to fall on to a sheet metal platform fixed to the machine upright and so arranged that it overlaps and is slightly clear of the machine table. A scraper is fixed to the sloping guard, and as the table advances, this scraper pushes the swarf along in front of it. As the swarf passes clear of the machine upright it falls into a trough on the side of the

machine bed. This arrangement is, of course, effected on both sides of the machine. The trough has an opening low down on the side for the easy removal of the accumulated swarf.

After the milling operation is completed, the block is lifted by electric hoist and deposited on the starting point of the roller conveyor. The initial milling operation provides two datum faces for subsequent operations, but they merely give vertical location and it is necessary to have both transverse and longitudinal location points also. They are provided at the next operation when two holes are drilled and reamed in the sump face on an Archdale 3ft. 6in. light radial drilling machine. Another hole is drilled, counterbored and a seat formed and the sides of the distributor boss are cleared on the same machine.

From the radial drilling machine, the block progresses to a Cincinnati special vertical two-spindle milling machine on which the bearing cap seats and tenons and side faces for end caps are machined. An allowance is left on the seats and tenons to be removed by broaching at a later operation. The set-up for this operation is shown in Fig. 2. Two 5in. diameter Galtona cutters with 16 teeth are used. The cutting speed is 240ft./min. and the tooth load 0.0035in. for each cutter.

Another Cincinnati milling machine is used at the next operation. It is a six-spindle special duplex machine on which the fuel pump face, the engine number pad, two angular faces, one on each side, the tappet cover and core cover facings and the top of the distributor boss are all machined at the one setting. Galtona cutters are used on all spindles and the machining conditions are :—

Cutter dia. inches	No. of teeth	Speed ft./min.	Tooth load inches
7	14	300	0.0035
$4\frac{1}{2}$	10	430	0.0021
$3\frac{5}{8}$	8	430	0.0020
$4\frac{1}{2}$	6	380	0.0045
$5\frac{3}{4}$	14	400	0.0020
$4\frac{3}{4}$	12	375	0.0021

Because of the type of operation, it is not possible to use an open end fixture with this machine. Therefore some transverse movement of the work is inevitable. To reduce this movement to a minimum the machine is so placed that the table runs alongside the roller conveyor as may be seen from Figs. 3 and 4. To facilitate movement of the block in to and out of the work fixture there are two rows of five small rollers mounted in the base of the work fixture, one at one end and one at the other. These rollers are eccentrically mounted so that they have a high position and a low position. In the higher position they stand slightly higher than the locating pads on which the block rests for machining so that only very low rolling friction has to be overcome in loading or unloading the fixture.

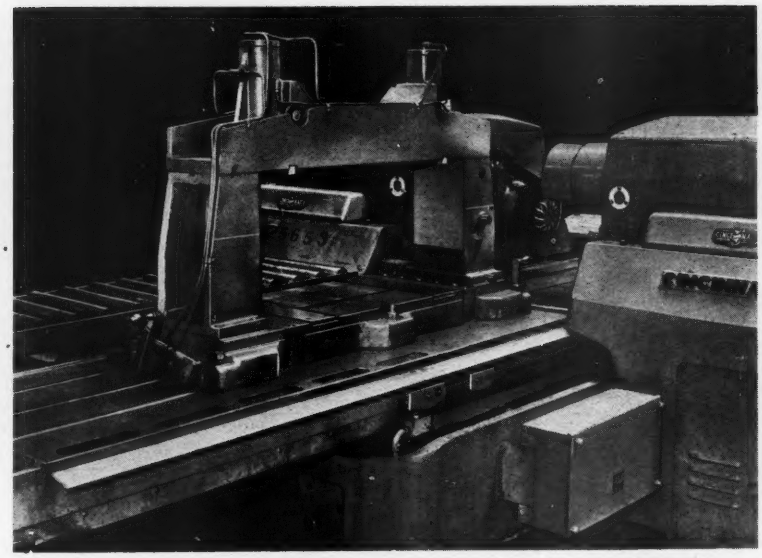


Fig. 3. The work-holding fixture in the loading position for the machine shown in Fig. 4.

As the block is loaded into the fixture it contacts two stops that give approximate location. Movement of a small lever at the end of the fixture then lowers the rollers to allow the block to rest on the base of the fixture. Movement of this lever also advances two plungers to register in the reamed dowel holes and give accurate location. The same lever also controls a valve in the hydraulic clamping circuit and until the block is correctly positioned, clamping cannot be effected nor can the table traverse be started until the clamps are properly applied.

At the next operation the main oil feed hole is drilled in an Archdale duplex machine. Once again, the operation is of a character that calls for transverse movement of the cylinder block in relation to the conveyor

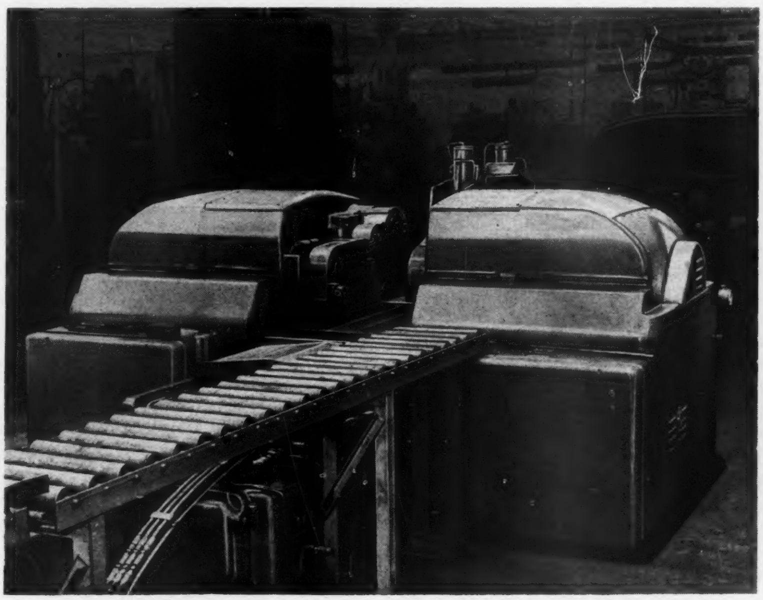


Fig. 4. Cincinnati duplex machine for milling both sides of the cylinder block.

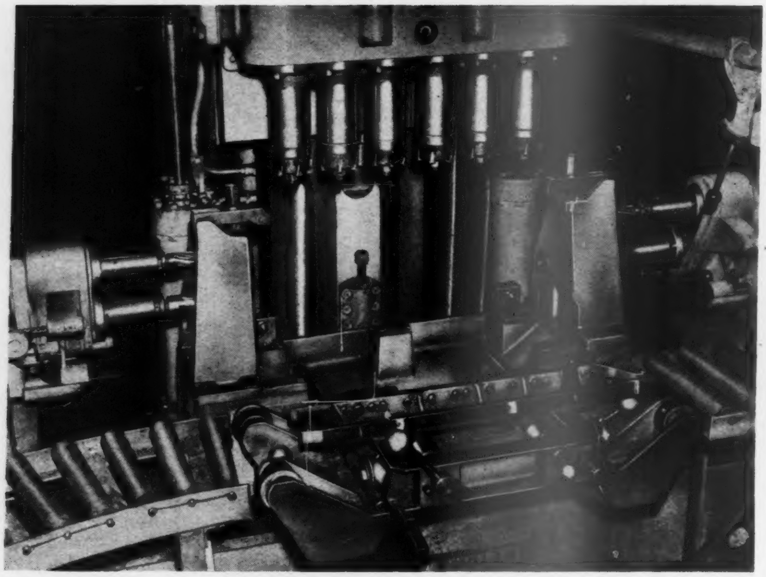


Fig. 5. A work fixture with traversing undercarriage.

line. To facilitate this movement the work-holding fixture incorporates an under-carriage that can be traversed at right angles to the line of the conveyor. At its forward position the under-carriage is in line with the conveyor and it is held in this position by a retractable stop. Four fairly large diameter rollers are eccentrically mounted in the under-carriage so that for loading and unloading the cylinder block can be lifted clear of the pads on the fixture.

As the block is pushed from the conveyor into the fixture, approximate transverse location is taken from the bearing tenons and approximate longitudinal location from a lever-operated stop. When the approximate location is reached, movement of a ball-end lever lowers the rollers and allows the block to rest on the fixture. The under-carriage with the block is then pushed into position on the machine. This is an easy matter since the undercarriage is mounted on rollers running on ball bearings. When the undercarriage reaches a back stop it is automatically locked in position. The dowels to give correct location are then advanced and the block is clamped ready for machining. This type of fixture is most effective in allowing movement of a heavy component with little physical effort. An undercarriage fixture is shown in Fig. 5, but it should be understood that this illustration does not refer to the duplex drilling machine.

After the main oil hole has been drilled, the block is transferred to a Cincinnati horizontal broaching machine in which bearing cap seats and tenons, part milled at an earlier operation, are finish broached to size. The next four operations in the sequence are all carried out on Archdale machines with multi-spindle vertical heads and two independent horizontal heads parallel to the conveyor line. The horizontal heads are arranged one on either side of the vertical head so that one machines the front face and the other the rear face of the block. They are all standard cam-feed units. Since these horizontal heads are arranged to work on the end faces of the block, it is impossible to use open end work-holding fixtures. For this reason a work fixture with a traversing undercarriage of the type described in connection with drilling the main oil hole is used with each machine.

At the first of these four operations the cylinder bores are rough bored from a six-spindle vertical head and at the same time two plug holes, one in each end face, and the front and rear camshaft bores are core drilled from the horizontal heads. There is heavy stock removal at the cylinder boring operation and in order that this may be carried out satisfactorily snout type spindles of special design are incorporated in the machine. Galtona cutters with 6 teeth are used at a cutting speed of 200ft./min. and a tooth load of 0.0027in. Fig. 5 shows the work fixture undercarriage advanced to receive a block, and Fig. 6 shows the block in position ready for rough boring.

The next machine in this sequence has 16 spindles in the vertical head. They are used to core drill 14 holes in the cylinder head joint face to ensure clearance for bushes used on the operation of drilling the cylinder head holding down bolt holes, and for drilling two holes that are reamed at a later stage to take the locating dowels in the cylinder head. At the same setting the two horizontal heads are used for counterboring the rear cam bore, spot facing the oil gallery and drilling a hole in the rear face and spot facing the core plug hole in each end face.

On another Archdale machine of similar type, the 14 cylinder head

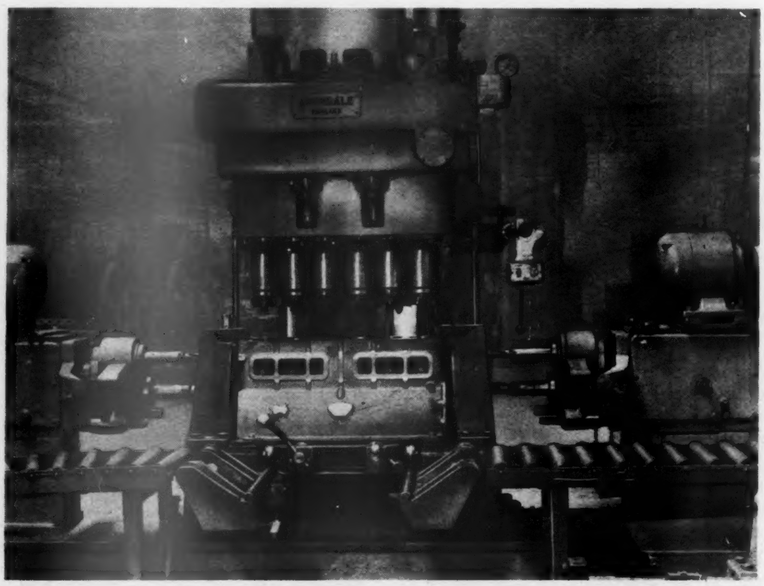


Fig. 6. The traversing undercarriage and the block in position for rough boring the cylinder bores on an Archdale six-spindle vertical borer with two independent two-spindle horizontal heads.

holding down bolt holes are drilled from a vertical head and the horizontal heads are used for core drilling the intermediate cam bores. At the final machine in this sequence the cylinder head bolt holes are reamed from the vertical multi-spindle head and the plug holes in the end faces are reamed from the horizontal spindles. At this stage a percentage check is made on the accuracy of the work. As only a percentage check is necessary the inspection station is not incorporated in the conveyor line. Instead, blocks to be checked are transferred by electric hoist to a short length of independent roller conveyor for moving to the inspection fixture shown in Fig. 7, which is designed to allow all relevant dimensions to be checked quickly and accurately.

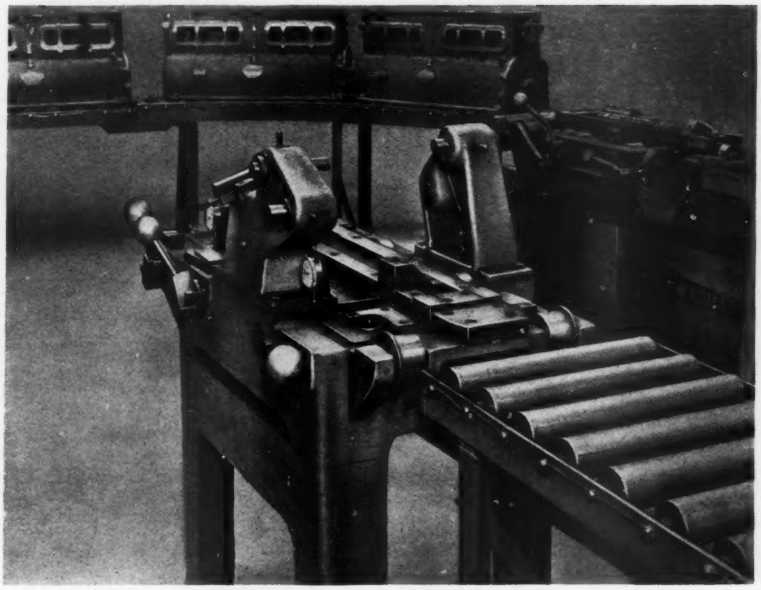


Fig. 7. An inspection fixture for part-machined cylinder blocks.

The next operation is carried out on an Archdale duplex multi-spindle drilling machine, see Fig. 8, on which all the holes on both sides that are normal to the axis of the block are drilled. In all, 85 holes are drilled at this setting. These holes are countersunk with portable tools and the block then passes to an Archdale multi-spindle duplex tapper in which the holes drilled at the previous operation are tapped. This machine also incorporates two drilling heads for reaming the location holes in the joint face. After the block leaves this machine, plugs and covers are fitted and the block is water tested under pressure, see Fig. 9. Every block is given this test, and therefore the inspection station is incorporated in the conveyor line.

After inspection, the block is transferred by electric hoist to a special Archdale two-spindle horizontal boring machine with an independent single-spindle horizontal drill head. On this machine the camshaft bores and the half-bores of the crankshaft bearings are rough bored. To machine half-bores is perhaps against normal practice since it entails interrupted cuts with a possibility of damage to tools, but experience has shown that it is perfectly satisfactory. The independent single-spindle head is used for drilling in the fuel pump face while the boring is being carried out.

The three succeeding operations do not call for detailed comment. Two are drilling operations carried out on Archdale vertical multi-spindle machines with angular attachments and the third is carried out on a six-spindle Archdale machine on which the bores to take the cylinder liners are rough counterbored. It may, however, be remarked that extremely open, and easily loaded and unloaded, fixtures are used at all three machines.

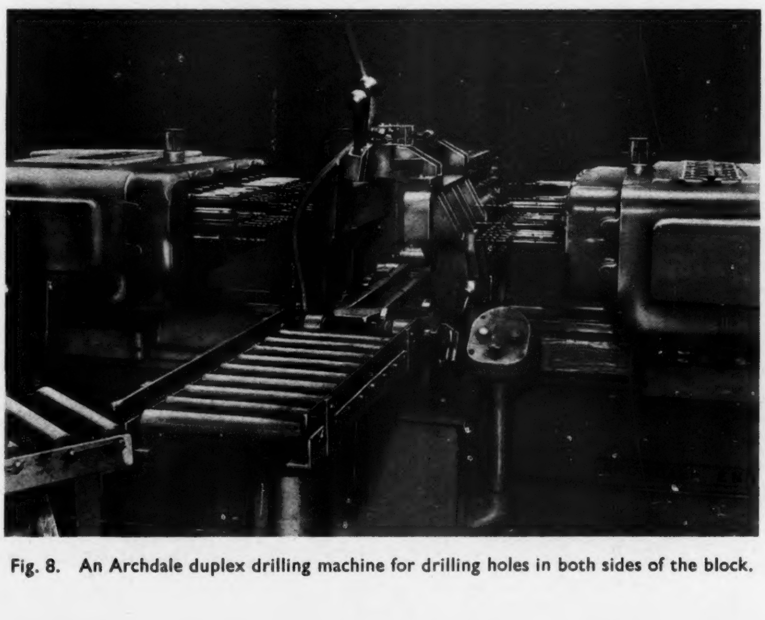


Fig. 8. An Archdale duplex drilling machine for drilling holes in both sides of the block.

A special Archdale horizontal milling machine is used at the next operation. There are 20 Galtona cutters mounted on this machine for milling the rear end bearing outer face, the sides of all crankshaft bearings and the lock tag slots. If all these faces were to be milled at one setting on a conventional machine, loading would entail mounting the work on a face plate, a lengthy procedure. To obviate this, James Archdale and Co., Ltd., have developed the machine shown in Fig. 10. It is built from elements of a standard drilling machine and a standard horizontal milling head. On this machine the block can be pushed direct from the conveyor into the machining position.

The cutter for milling the rear end bearing outer face is 8in. diameter and has 12 teeth. It runs at a cutting speed of 147ft./min. and a tooth load of 0.0016in. Six right-hand and six left-hand cutters are used for milling the sides of the crankshaft bearings. They are all 5in. diameter with 14 teeth and run at a cutting speed of 92ft./min. and a tooth load of 0.0013in. For milling the lock tag slots there are seven cutters, all 3in. diameter with 16 teeth, running at a cutting speed of 55ft./min. with a tooth load of 0.0011in. From this machine the block is transferred to an Archdale six-spindle vertical borer for semi-finishing the cylinder bores.

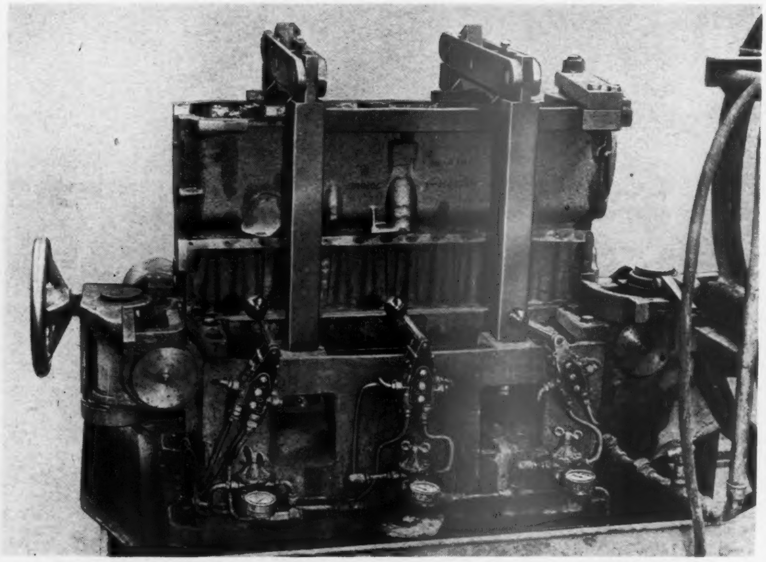


Fig. 9. A block on pressure water test.

An interesting work-holding fixture and an unusual machine cycle are used at the next operation, at which the push rod holes are drilled. There are 12 holes to be drilled, but the inlet and exhaust for each cylinder are at such close centres that it is impossible to mount 12 spindles to drill all the holes in one traverse of the machine head. They are drilled in two traverses on a six-spindle Archdale machine with a special cycle. This machine with the block in position in shown in Fig. 11.

As an open end fixture is used, the block is merely pushed along the conveyor and into the fixture. There are rollers, eccentrically-mounted in the base of the work fixture, to facilitate movement of the block into position. The block rides over these rollers until it contacts a retractable stop that gives approximate location. At this stage an air-operated mechanism is brought into action to lower the rollers and advance the dowels to give accurate location; and two air-operated clamps are applied to hold the work securely.

The work fixture can take up three positions on the machine bed in relation to the machine spindles. In the first, or loading position, the fixture is centrally under the machine head. In this position none of the points to be drilled is in line with a spindle. Of the other two positions for the fixture, one is slightly to the left and the other slightly to the right of the first position, so that a slight movement one way brings the inlet holes under the spindles, and a slight movement the other brings the exhaust holes under the spindles.

It is necessary to have three positions for the work fixture to allow the drill bush plate to enter the block, since the space available is so tight that the drill bush plate ends are turned to allow the bush plate to enter the camshaft bore when the fixture is moved to bring one or other set of holes under the drills. Initial fast traverse of the head from the

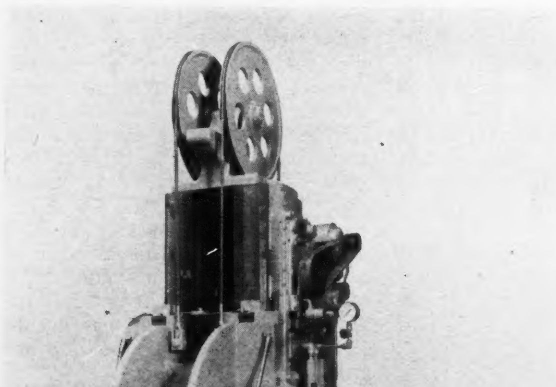


Fig. 10. A special Archdale milling machine with 20 cutters for machining bearing faces and lock tag slots.

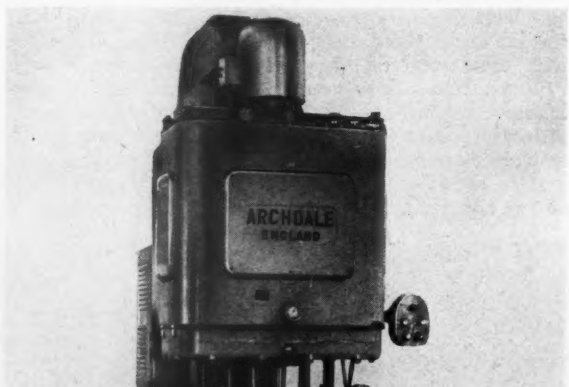


Fig. 11. Set-up for drilling the push rod holes. The machine has a special work cycle.

fully retracted position can be effected only when the fixture is in the central position. This fast traverse continues until the drill bush plate is in line with the camshaft bores. The downward travel of the head is then stopped automatically and the work fixture is moved to bring one set of holes in line with the machine spindles. It is then possible to start the automatic working cycle of fast approach, feed and retract to the position at which the initial fast approach stopped.

The fixture is then moved to the other working position and the cycle repeated. Movement of the fixture is effected by means of a lever and there are stops to give exact location. At the end of the second cycle the head retracts to the position at which the working cycle begins. The fixture is then once again brought into the central position and the head is fully retracted. As soon as the fast upwards traverse of the head is completed, the clamps are released, the dowels are retracted and the rollers raised. It is then only necessary to retract the approximate location stop by a simple movement of a lever and the cylinder block can be pushed out on to the conveyor for transfer to the next machine.

After the push rod holes are drilled, the block passes through four operations on Archdale angular and vertical drilling machines. These are conventional operations and do not call for special comment, although at the last machine in this sequence 14 holes are drilled in the bearing cap seatings and 26 in the sump flange. The next operation is interesting, inasmuch as it may be described as manually-operated transfer machining. A single fixture holds the work successively under an Archdale two-spindle tapping head and two Archdale single-spindle drill heads, see Fig. 12. The holes to be drilled are at such close centres that it is impossible to deal with them simultaneously. For this reason, the work is first brought into position under the tapping head, then traversed with the fixture into position under the first drill head, and finally into position under the second drill head. The design of the fixture and the traversing mechanism is such that at each stage the correct position is obtained with any dependence upon the skill of the operator. This arrangement has two advantages. It allows three operations to be carried out for one loading and it also gives a time for this station that is reasonably in agreement with the times at other stations.

Two drilling operations on an Archdale four-spindle and an Archdale single-spindle machine and a tapping operation on an Archdale multi-spindle tapper on which 41 holes are tapped simultaneously, follow. After these, the block is progressed to an Archdale six-spindle machine for reaming the push rod holes. This machine and the work fixture are similar to, and operate in the same manner as, the machine and work fixture used for drilling the push rod holes, so that all 12 holes are reamed for one loading.

At this stage the bearing caps are assembled to the cylinder block. The machining section for bearing caps is laid out in such a manner that the last operation on the caps as individual components is completed adjacent to the station in the cylinder block line at which assembly takes place. A description of the methods used for machining the bearing caps is given later in these notes.

As soon as the bearing caps have been assembled, the block passes to an interesting Archdale combined duplex milling machine, see Fig. 13. One head of this machine carries a single

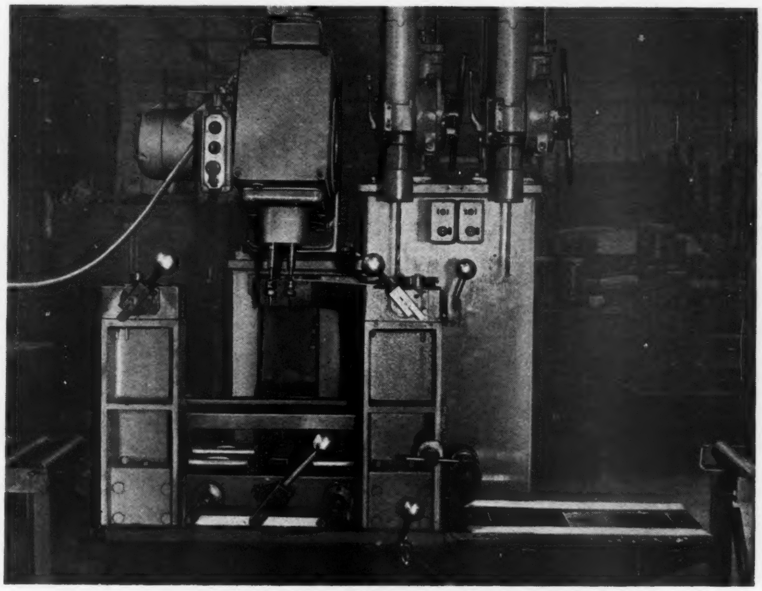


Fig. 12. Three station fixture used in conjunction with a two-spindle tapper and two single-spindle drilling heads.

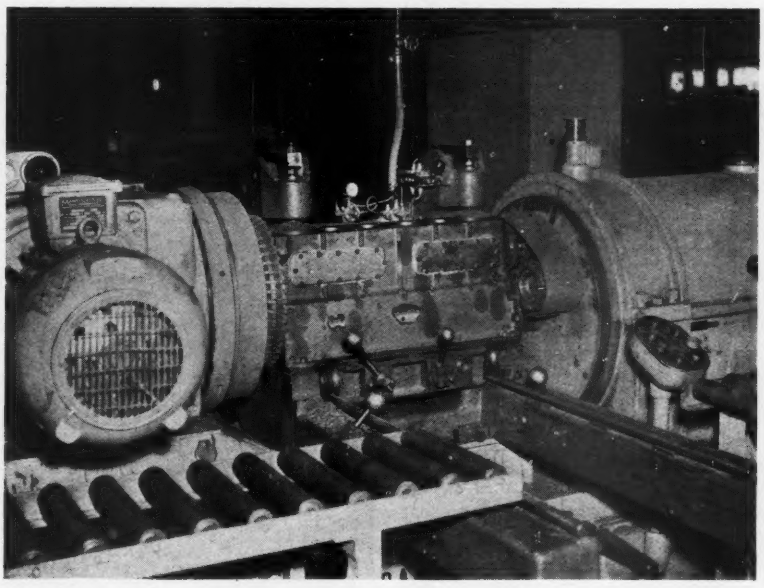


Fig. 13. Milling the end faces simultaneously on an Archdale combined plain horizontal and planetary milling machine.

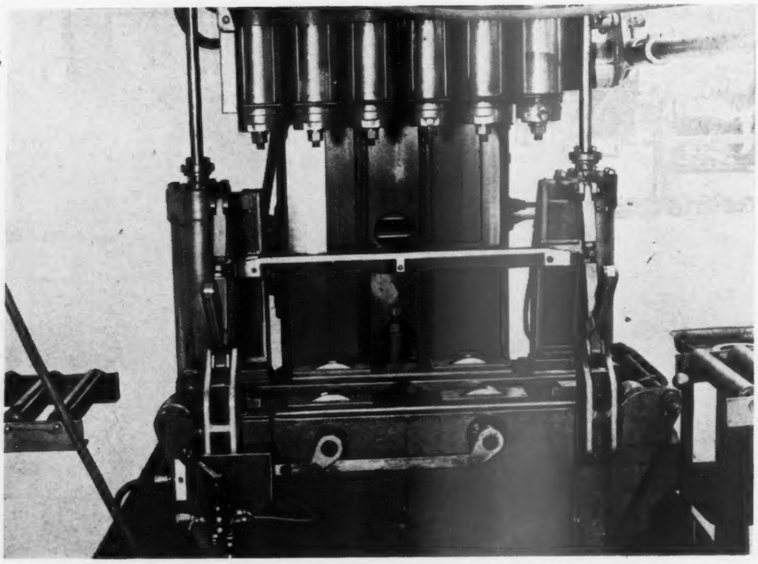


Fig. 14. The work-holding fixture on the machine for fine boring the cylinder bores.

18½in. diameter cutter with 50 teeth. This is for finish machining the front end face of the block and bearing. It runs at a cutting speed of 225ft./min. with a tooth load of 0.0056in. The depth of cut is 0.125in. At the same time an Archdale planetary milling head machines the rear face. This head carries a roughing and finishing cutter. Both are 6in. diameter and have 17 teeth. For the roughing operation the cutting speed is 103.5ft./min. with a tooth load of 0.0057in. and 0.125in. depth of cut. The corresponding figures for the finishing cutter are 157ft./min., 0.0031in. and 0.020in.

From the milling operation the block passes to an Archdale three-head horizontal multi-spindle drilling machine on which holes are drilled in the front and rear faces and in one side. Between this machine and the next, the tapping holes are countersunk; a portable tool is used. The block is then transferred to an Archdale four-way multi-spindle tapping machine, on which holes for the cylinder head studs, in the front and rear end faces and in one side are tapped.

For the next operation the block must once again be lifted off the conveyor by electric hoist for loading in to an Archdale two-spindle horizontal boring machine in which the complete crankshaft bearings are semi-finished and the camshaft bores are finish machined ready to receive the white metal camshaft bushes. This machine is similar in character to that used for the initial boring of the crankshaft half bearings and the camshaft bores. Two holes are then tapped on a two-spindle machine, after which the block is again inspected.

Before any further machining is carried out, all bearing caps are removed. They are, of course, stamped for identification to ensure correct re-assembly. The next machining operation is carried out on a special Archdale horizontal milling machine that carries two cutters for milling an oil slinger groove in the rear bearing of the block and in the rear bearing cap. From this machine the block passes to an Archdale duplex drilling machine which is tooled for finishing a counterbore undercut and for facing the centre bearing thrust faces to width. The centre bearing cap is then removed and two slots are milled in it and in the rear bearing cap on a Cincinnati plain auto mill.

For the important operation of fine boring the cylinder block to size for the cylinder liners, an Archdale six-spindle machine shown in Fig. 14 is used. As can be seen from the illustration, an extremely open type of work fixture is used so that loading and unloading are very easily effected. The air cylinder control mounted at the right-hand side of the machine base actuates both the locating dowels and the eccentrically-mounted rollers on which the block rides when it is being pushed into, or out of, the fixture. Clamping is effected manually by means of a quick-action clamp at each end of the fixture.

Single point tools are used for this operation, and an interesting feature of the set-up is the manner in which provision is made for withdrawing the tools without scoring the machined bore. At the end of the feed traverse, by simple movement of a lever on the machine, the operator ensures that all the spindles come to rest with the tools in the same position, namely, at the back. The circuit for the machine drive is broken. Movement of another lever then moves the work holding fixture and the block a small distance towards the back of the machine, sufficient to bring the tools clear of the machined bores. This movement brings the fixture into contact with a limit switch which closes the circuit for the machine drive and rapid withdrawal of the head takes place automatically.

From the fine boring operation the block is passed to an Archdale H.D. vertical milling machine, see Fig. 15,

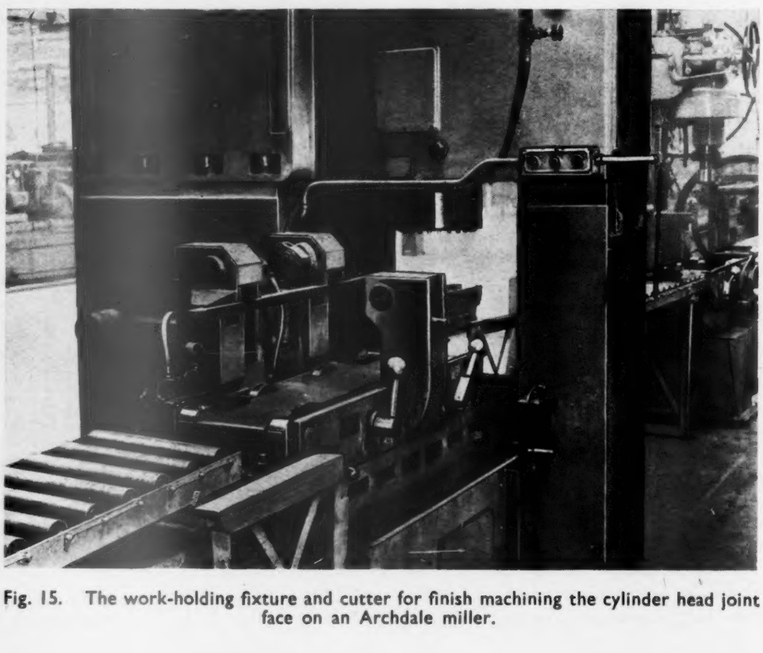


Fig. 15. The work-holding fixture and cutter for finish machining the cylinder head joint face on an Archdale miller.

on which the joint face is finish machined. For this operation a Carbifine 10in. diameter cutter with 24 teeth is used. The cutting speed is 393ft./min. and the tooth load 0.0056 in. On yet another Archdale machine, the counterbores for the cylinder liners are finished and the top end of each bore is undercut and chamfered.

After a frazing operation, the camshaft bushes are pressed into place and the bearing caps are re-fitted and the block is ready for the final machining operation. This is carried out on an Archdale two-spindle horizontal machine tooled for fine boring the camshaft bushes and the crankshaft bearings. It is similar in type to the machines used for the earlier boring operations on these elements of the block. This machine also incorporates an independent small two-spindle horizontal drilling head used for reaming two dowel holes in the rear end facing to give accurate register for locating the clutch housing.

From the final machining operation the block passes to a washing machine in which it is thoroughly cleaned. When the washing is completed, the case and all oil passages are blown out with compressed air to ensure that there is complete freedom from swarf and other foreign matter. At the same station on the conveyor all the oil holes are frazed. Before the cylinder liners are fitted the block is given a complete inspection for dimensions and finish. To facilitate this inspection the block is mounted in a turn-over device that allows it to be turned easily to bring through-holes into such a position that lights mounted behind the inspection station make it an easy matter to ensure that these holes are perfectly clear and of the necessary standard of finish

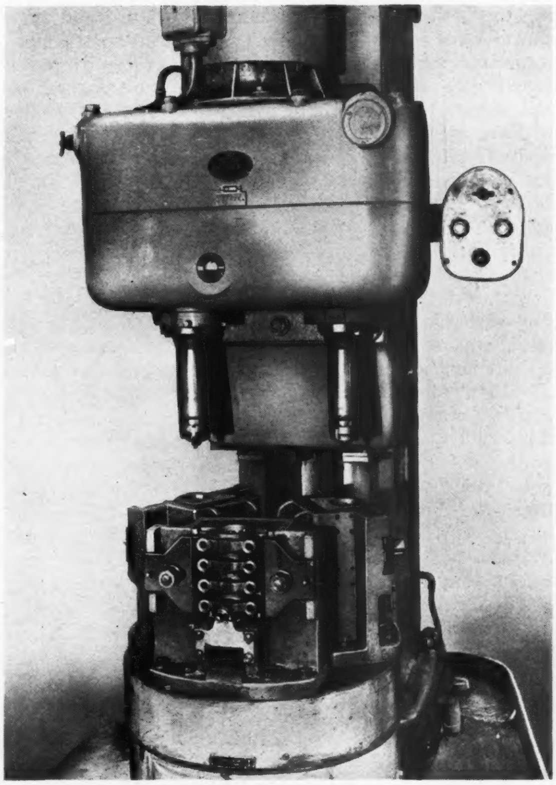


Fig. 16. The set-up for rough and semi-finishing the half-bores of composite bearing cap castings.

Crankshaft Bearing Caps

All seven crankshaft bearing caps are produced from two castings, one for the front centre and rear caps and the other for the intermediate caps. The major machining operations are all carried out before the castings are split into individual bearing caps. Since in the one case four and in the other three caps are machined simultaneously, this use of composite castings considerably reduces the time spent in loading and unloading machines to produce a specified number of caps.

The initial machining on both sets of castings is performed on a Weatherley Oilgear double-slide vertical surface broaching machine with a normal broaching capacity of 30,000lb. and a stroke of 66in. This is a standard machine with two automatic shuttle tables. As one table moves into the broaching position the other simultaneously moves out. At the conclusion of the table movement, one slide automatically moves down and at the same time the other moves up to the starting position. Neither tool slide can start its downward travel until its shuttle table is hydraulically and mechanically locked in the broaching position. Nor can either slide move up until its shuttle table is in the unloading position. To give positive location and simple yet accurate adjustment, hardened stop blocks on each shuttle table contact pre-set micrometer screws when the broaching position is reached.

Fig. 17. Checking a cylinder liner on a special Solex gauge.

In operation, work is loaded in the left-hand shuttle fixture and dual push buttons are depressed to start the machine cycle. These push buttons are so placed that the operator must use both hands to depress them, and as they must be depressed simultaneously they give complete operational safety. When the machine cycle is started, the left-hand shuttle table moves in to the broaching position and is hydraulically and mechanically locked in place. At the same time the right-hand shuttle table moves out to the unloading and loading position. Therefore, while one slide is broaching, the operator unloads and loads the work fixture for the other slide and there is no lost time. The tool slides stop automatically when the pre-set stroke is completed. Further depression of the push buttons then starts another cycle with the right-hand shuttle moving in to, and the left-hand moving out from, the broaching position.

For broaching the crankshaft bear-

ing caps, the work fixture on each shuttle table is arranged to make two composite castings. These fixtures have been so designed that both three and four-part castings can be broached without any change of tools. If so desired, batches of both types can be run through together.

At the first station on the left-hand table the casting is automatically pulled back against the bosses by hook clamps applied in the half bore for the joint face to be broached to size. A progressive broach is used for the initial cuts and finishing is effected by a short length of slab broach. When the shuttle table retracts to the loading position, the casting is transferred to the second station in the work fixture on the left-hand table. At this station it is loaded with the joint face away from the broach and acting as a location surface for the back faces to be broached to give the correct cap height.

The component is then transferred to the work fixture on the right-hand shuttle table. This also has two stations. To ensure that when for any reason a bearing cap is removed after it has been fitted to the cylinder block it must be refitted in its original position, the cap is non-symmetrical about the axis of the bore. For this reason, the work-holding arrangements at the third and fourth stations incorporate simple but effective provision to ensure that the fixture is correctly loaded. At the third station the bearing cap sides are broached to width. Once again, the initial cuts are taken with a progressive broach, but for the final cut an adjustable slab broach is used, since the dimensional tolerance is small. This completes the broaching on the castings for the intermediate bearings. Castings for the front, centre and rear bearings proceed to the fourth station where an oil seal groove is produced in each side of the front and rear caps.

After they have been broached, the castings are transferred to an Archdale horizontal multi-spindle drilling machine. The work fixture on this machine is arranged to take one of each type of casting so that once again there is balanced output. One head carries 14 drills for drilling the bolt holes halfway through. The other head carries 14 similar drills for drilling the bolt holes halfway through from the other face and 8 smaller drills. The heads feed in simultaneously. At a pre-determined point the back head is automatically retracted while the front head continues to advance until its drills break through into the holes part drilled from the rear head.

Minor operations are carried out on a single spindle vertical drill and a small radial drill before the next major operation. This is carried out on an Archdale two-spindle vertical borer. A three-station indexing fixture is mounted on the table of this machine. Each station on the work-holding fixture is arranged to take one of each type of casting. The two castings are clamped together so that the tools work in a full bore. At the first working station the bores are rough machined and at the semi-finished. Once again, of course, there is perfectly balanced production.

Lock tag slots are then milled on a Cincinnati plain auto mill, and the castings are ready to be split into individual caps. The splitting operation is carried out on a Cincinnati plain hydro mill. Two work holding fixtures are mounted on the machine table. They are diagonally opposed, one at each end. The fixture at one end takes the intermediate cap casting. For machining this, there are five milling cutters, namely two end mills for machining the outside faces and three slitting cutters. At the other end the fixture takes the rear, front and centre cap casting. For machining this there are three milling cutters, namely an end mill for machining the outside face of the rear cap and two slitting cutters for parting the casting into individual caps. It should perhaps be explained that there is no need to machine the outer face of the front cap at this stage, because this face must be machined on the combined horizontal and planetary miller in the cylinder block line immediately after the bearing caps are fitted to the block. This completes the machining on all except the rear and front caps which pass through minor drilling operations before transfer to cylinder block line.

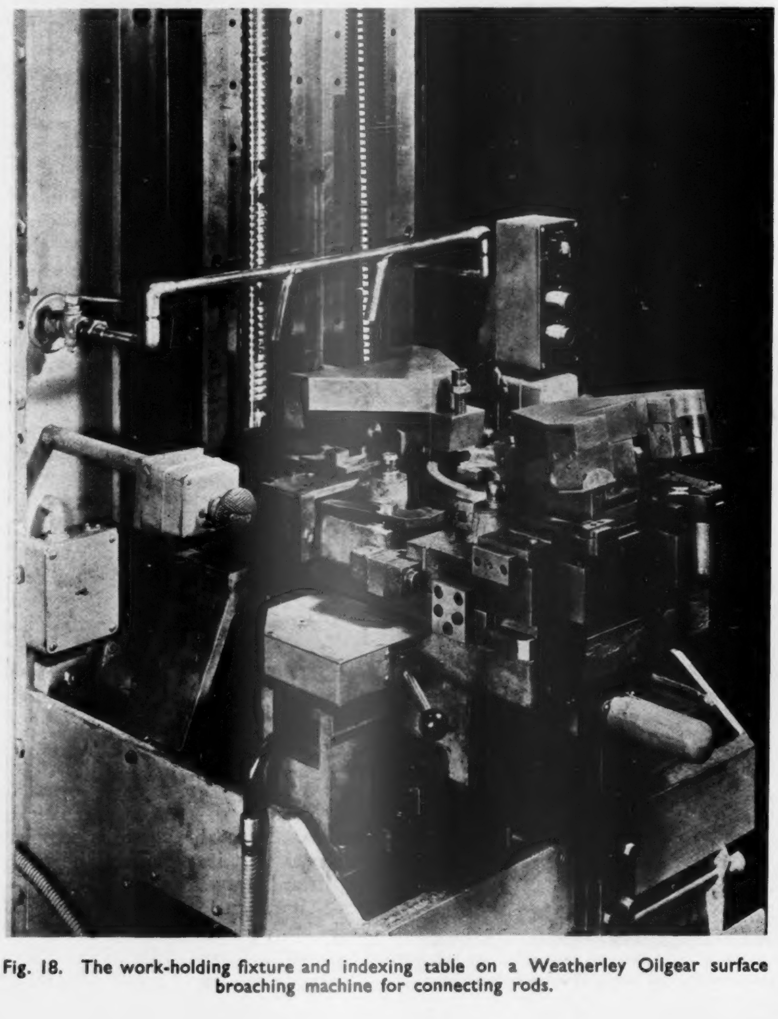


Fig. 18. The work-holding fixture and indexing table on a Weatherley Oilgear surface broaching machine for connecting rods.

Cylinder Liners

Production methods on the cylinder liners follow conventional lines. Although they do not call for detailed description, the sequence

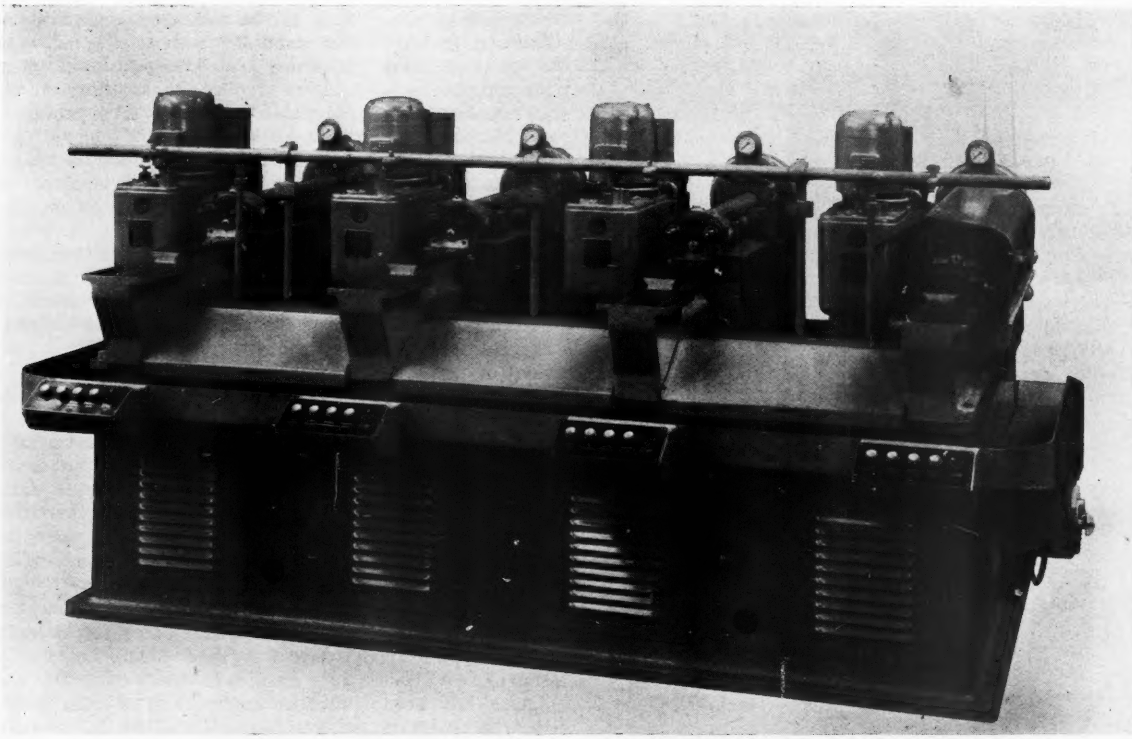


Fig. 19. Special Archdale deep hole drilling machine. Eight connecting rods are drilled simultaneously.

adopted to give quick and accurate work may be briefly discussed. At the first operation on a Maximinor multi-tool lathe both ends are faced and chamfered and the O.D. of the manufacturing flange is turned. A drive slot is then milled in the manufacturing flange, following which the barrel diameter and the top diameter of the flange are rough turned in a second Maximinor multi-tool lathe.

Rough boring, which is the next operation, is carried out on a six-spindle Archdale vertical cylinder borer. After this the bore and the barrel diameter are semi-finish machined simultaneously on a Heald Borematic that takes three liners at a time. At the same setting the top diameter of the liner flange is also semi-finished. From the Borematic, the liner is transferred to a Herbert senior 2nd operation lathe in which the small end is faced to length and parted off from the manufacturing flange.

At this stage the liner is ready for the series of operations that produce close dimensional accuracy and a very high degree of surface finish. At the first the bore is fine bored on a Heald Borematic that takes three liners at a time. An allowance is left for honing. Following this and on a similar Heald machine, the barrel diameter is finish turned with an allowance for grinding, and the top diameter of the flange is finish turned. At two succeeding operations the small end is chamfered inside and out, and the flange is straddle faced and undercut and the bore chamfered. To complete the machining, the barrel diameter is ground on a Precimax fine grinder and the bore is honed on a Barnes machine. Before transfer for assembly into the cylinder block, every liner is thoroughly examined for quality of surface and for dimensional accuracy. Diametral accuracy, both internal and external, is checked on a special Solex gauge, see Fig. 17.

With the Solex gauge a liner can be checked in four seconds. By means of 36 tiny air jets diametral measurements are taken simultaneously at six places without there being any contact between the liner and the gauging elements. As the liner does not take its true shape until it is fitted into the cylinder block, the readings are of the average sizes at top, centre and bottom, both external and internal. They show what the size will be after fitting. The two larger illuminated windows either side of the plunger give the sizes instantaneously. Other tell-tale dials and windows constantly show at a glance that the gauge itself is functioning correctly. The operator merely places the liner under a hydraulic plunger, then movement of a lever causes the liner to be pushed down into the gauging position. Release of the lever extracts the liner after it has been checked. Measurements are made to 0.00002in.

Connecting Rods

There is a particularly compact lay-out for the connecting rod section, that at first sight looks to be haphazard rather than planned. This impression arises from the fact that the machines are not laid out in orderly rows. However, closer consideration shows that the apparent lack of order conceals a cleverly conceived lay-out that makes the best use of the available floor area and also carries the work forward in an orderly sequence to the sub-assembly section adjacent to the engine assembly conveyor.

Connecting rods are produced from separate stampings for the rod and the cap. Before the rod stampings are delivered to the machining section the big end boss is coined to leave a small machining allowance, and the small end boss and the girder are coined to width. The first machining operation is carried out on an Archdale six-spindle vertical drilling machine that carries two drills, two rough reamers and two finish reamers. On this machine the small end is drilled, rough and finish reamed. The work fixture is of the indexing type. It has four stations, three working and one for loading and unloading. Two rods are mounted in the work fixture at each station. On leaving this machine the small end hole is chamfered on a Herbert single-spindle sensitive drill.

A Weatherley Oilgear single - slide vertical surface broaching machine is

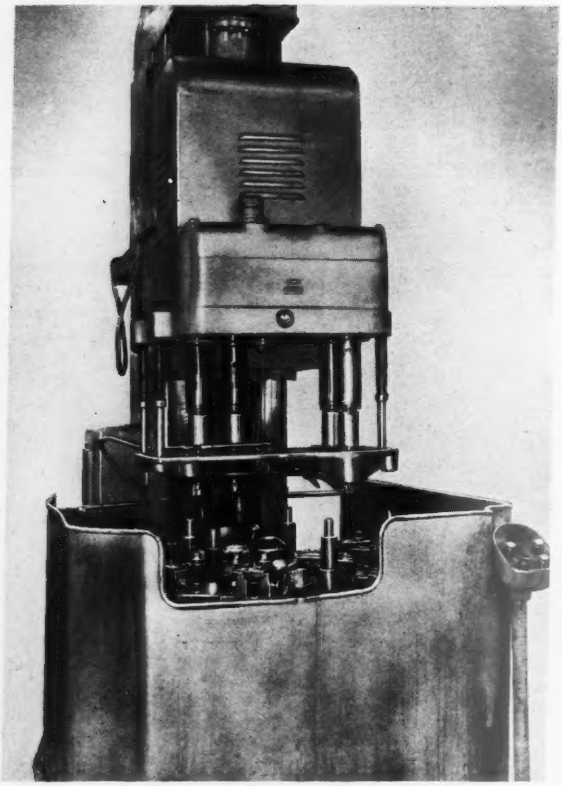


Fig. 20. An Archdale machine for rough and semi-finishing the bore of the connecting rod big end

used at the next operation. In essentials this machine is similar to the Weatherley machine used for broaching the crankshaft bearing caps, except that it has only one slide. Although there is only one slide the work fixtures and the broaches are designed to allow two broachings to be carried out simultaneously on one connecting rod. The work cycle is that at the first station the half-bore is broached to a size that leaves an allowance for the finish machining operations, and the sides of the bolt hole bosses are broached to size. At the second station the joint face and the backs of the bolt bosses are finish machined.

A two position indexing fixture is used which allows loading at one station while broaching at the other. It is shown in Fig. 18. Location for the first station is taken from the reamed small end bore and from the outer diameter of the big end. The location at the big end is designed to centralise the half-bore. It is effected automatically by means of a vee plate operated through a cam-actuated lever that advances the vee plate as the work comes into the broaching position and retracts it as the table is indexed after the completion of a cycle. At the second station location is taken from the small end bore and the half bore broached at the first station.

A standard indexing fixture is mounted on the shuttle table, but the work-holding fixtures are specially designed for this application. The central cam, which can be seen in Fig. 18, not only actuates advance and retraction of the vee plate but also opens and shuts the automatic clamps. Indexing is effected manually and the correct table position is given by a spring loaded plunger in register with a vee in the indexing table. A limit switch device has been incorporated in the design, and until the plunger is correctly positioned in the vee, it is impossible to start the machine cycle. Forward travel of the shuttle table to the pre-set broaching position is controlled by an adjustable micro-stop.

All the broaches have been specially designed for this specific application. The circular broach for the half-bore is splined to give reasonable chip form and to reduce the cutting force. Inevitably the splines leave protuberances in the bore. To remove these there are three short sections of broach with teeth set at an angle to act as shear teeth. If the alternative 'double-jum' method were used, it would necessitate nearly twice the length of roughing broaches. In order that full use may be made of the circular broach it can be turned through 180 deg. to bring fresh cutting edges into action.

Broached rods are placed into a basket standing at the top of a chute leading from the machine. When a basket has a full load the operator pushes it on to the chute to be carried direct to a washing machine. After the washing the bolt bosses are shot peened. At the next major operation the bolt holes are drilled, and rough and finish reamed on an Archdale eight-spindle vertical drilling machine. This operation is interesting inasmuch as the six station indexing table carries three fixtures for connecting rods and three for connecting rod caps, spaced alternatively. There are four working stations to give the following cycle :—

(1) Drill two holes halfway through.
(2) Drill through two holes.
(3) Rough ream two holes.
(4) Finish ream two holes.

The fifth station is for loading and unloading and the sixth is an idle station. By this method one rod, or one cap, is completed at each cycle of the machine. As both rod and cap are produced by the same tools accuracy

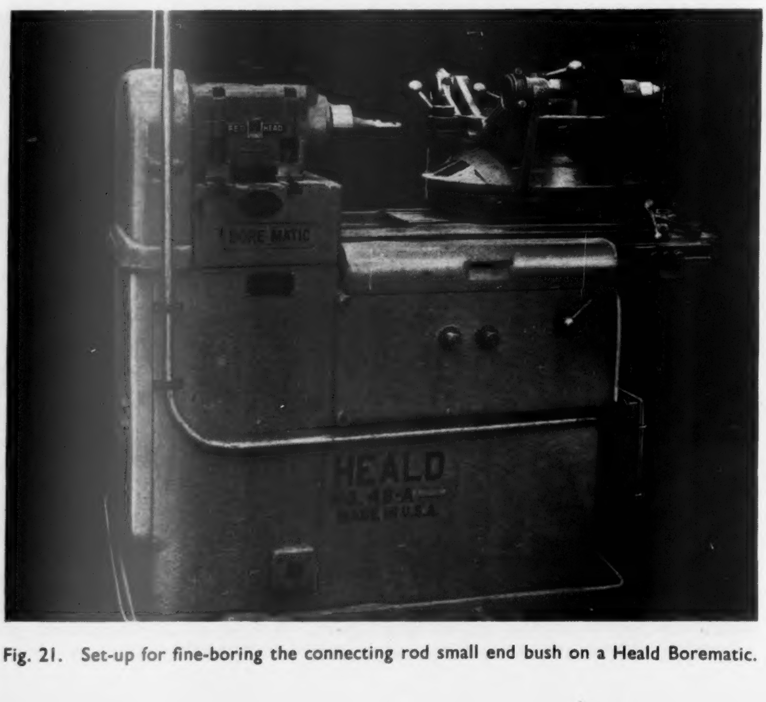


Fig. 21. Set-up for fine-boring the connecting rod small end bush on a Heald Borematic.

of size and centre distance is maintained. This is essential to give the dowelling effect required when the bolts are fitted.

At the next major operation the long oil hole is drilled from the big end to the small end. This is carried out on a machine that has been specially developed by James Archdale and Co., Ltd., see Fig. 19. The machine incorporates four standard horizontal automatic hydraulic reciprocating drilling heads. A two-spindle drill head is attached to each standard head so that eight rods can be drilled simultaneously. After the oil hole is drilled, only minor operations are carried out until the cap is assembled to the rod. The first machining operation on the cap is performed on a Cincinnati vertical surface broaching machine, on which the side faces, the joint faces and the sides of the bolt bosses are machined. Only one other operation on the cap as a separate component calls for comment. It is that at which caps and rods are drilled and reamed alternately as described earlier.

When the cap is fitted to the rod, the assembly is transferred to another Archdale machine in which the big end is rough and semi-finish bored. This machine is shown in Fig. 20. It has four spindles carrying two roughing and two semi-finishing tools. The work-holding fixtures, each taking two rods, are mounted on a three-station indexing table. Of the remaining operations it will suffice to say that the small end and big end are fine bored on a Heald Borematic, the small end to the size for taking the bush and the big end to a size that leaves a small finish grinding allowance. The big end is ground to size on a Heald Gagematic internal grinder and the small end bush is fine bored on the Heald Borematic shown in Fig. 21. Very close dimensional accuracy and a remarkably high standard of surface finish are specified for and maintained on both bores.

Although these notes deal only with part of the engine detail machining department, it must be said that every section of the department shows evidence of similar careful planning to give a high rate of productivity without imposing excessive strain on the works. In fact, although the productivity per man-hour in this new department is much higher than it was in the old department, the actual physical effort expended in the course of a day is much less.

THE GAS TURBINE

Some Comments on its Application to Road Vehicles

IN an appendix to the James Clayton Lecture delivered to the Institution of Mechanical Engineers recently, Air Commodore F. R. Banks discusses the gas turbine for road vehicles. The lecture deals with "The Aviation Engine", and in the following extract the appendix is reproduced in full.

The gas turbine is not an impossibility for the automobile; experimental engines are being designed and some are already under test, but the turbine does not appear to have advantages compared with the normal piston engine, for this purpose. It may well have useful application in certain large commercial and military vehicle types and, perhaps, for the long distance passenger coach.

In the first place, the most suitable size of gas turbine to give reasonable efficiency is better fitted to the needs of the large vehicle rather than the automobile—since it is more easy to build an efficient gas turbine of 250 s.h.p. than one of 50 or 100 s.h.p. Scaling the engine down to these comparatively low powers demands lengthy and expensive development to obtain the required efficiency of components such as the compressor, the combustion chamber and the turbine.

Smoothness of operation and lack of vibration, inherent in the gas turbine, are now so good in the piston engine that a change to the former on these grounds alone could hardly be justified.

The idea that the gas turbine will use "any old fuel" may not be confirmed in practice, in the relatively low-power-vehicle engine. While anti-knock or detonation value is unimportant, the fuel must be fluid at all temperatures likely to be encountered in cold climates; and it must also give clean combustion, so that frequent cleaning or decarbonizing of the combustion chamber is not required. This would seem to rule out the so-called boiler or heavy fuels, because of their relatively high freezing point and their ash and asphaltic contents, etc. Light diesel or gas oil, or kerosene, are, therefore likely to be used, but these may be in short supply if any large number of turbine-engined vehicles emerge on the highways.

To compete with the piston type of petrol engine, the thermal efficiency of the gas turbine cannot be much less than 25 per cent. Although the efficiency of the former is relatively low at part-throttle and with the constant accelerations and decelerations of the automobile in ordinary usage, that of the gas turbine is likely to be considerably less. The gas turbine is a constant-speed or "full-throttle" engine and, therefore, its efficiency falls off seriously at engine speeds much below the maximum. The provision of a heat exchanger may, however, help in this respect.

The gas turbine will also have to meet the competition of the very high-compression piston engine, the development of which has now been brought to a practical stage (General Motors Research of America [Kettering 1947; Campbell, Carris, and Withrow 1948]).

An advantage of the gas turbine for automobile propulsion is that it should be possible to dispense with the multi-speed gearbox; since the usual arrangement, of a free turbine separated mechanically from the turbine-compressor system of the engine, will provide its own very satisfactory torque converter. A low-speed gear, for very steep gradients, and a reverse gear, will probably be the only mechanical speed changes required. But a multi-speed box may still be necessary for the heavy road vehicle.

The vehicle gas turbine will have a centrifugal compressor, since the axial type is somewhat impracticable and expensive to produce. Therefore, the pressure ratio of a small two-stage centrifugal compressor is hardly likely to be much more than 5/1 and it will do well to give 74 per cent. adiabatic efficiency. The matter of turbine efficiency is the most important for fuel economy since 1 per cent. improvement in turbine efficiency will be equivalent to about 3 per cent. improvement in fuel consumption. Since the efficiency of the aviation turbine is about 87 per cent., a similar efficiency will be difficult to achieve with such a comparatively small turbine. In view of the low pressure ratio of the compressor and the probable limitations in component efficiencies, a heat exchanger will be required.

Since the gas turbine will pump anything between six and ten times more air than the piston engine, for equal power output, the problems of silencing the air intake and exhaust

disposal are not inconsiderable, although, perhaps, not insuperable.

If the gas turbine eventually challenges the piston engine in the automobile, as distinct from its use in the heavy vehicle, this may come about purely as the result of an urge to be different, and not because it shows any better economy than the piston engine. Peculiar as it may seem, engineers often follow fashion for fashion's sake, but it is to be hoped that any such change to the turbine will be made only on the grounds of sheer merit, before discarding the many decades of experience of the established piston engine.

Finally, if the gas turbine were to be used in the automobile, it would have to be dealt with as a "packaged power unit", because the average garage or motor engineering concern would be unable, for some time, to handle any of the major or even minor maintenance problems of this type of engine. This would involve a considerable extension of the supplier's manufacturing and servicing facilities, to ensure a cushion of power units to meet such contingencies.

A HEAVY DUTY REAR BOGIE

Details of the Kirkstall Double Reduction Design

THE axles of the Kirkstall heavy duty bogie fitted to the Thornycroft "Mighty Antar" tractor, illustrated on page 325 of the *Automobile Engineer* for Sept./Oct. 1950, are of the double reduction type providing an overall ratio of 14.35/1 (primary 5.4/1, secondary 2.66/1), the secondary reduction being of the epicyclic type which advantageously spreads the high output torque over a number of equally loaded teeth. They are designed to take care of a tractive effort at ground of 26,000 lb. per axle and are particularly interesting by reason of the unorthodox compounding of the epicyclic trains.

On the assumption that the trend of vehicle development would inevitably demand driving axles of greater capacity within restricted dimensions, the possibility was investigated of combining together the differential and reduction gearing, with the result that a simple, compact and robust design was evolved. It might be of interest to mention that this new development was barely completed on the drawing board when an ideal application for its use arose on the request from Transport Equipment (Thornycroft) Ltd. for an outsize driving bogie to meet the requirements of the Iraq Petroleum Company.

The broad principle underlying the new Kirkstall axle calls for an epicyclic system with four inter-reacting members. A normal train has three members, but if two trains are used, combining one member from each train on a common shaft to react against each other and rotate as one, there remain four members, each of which is balanced against the other three provided that to each correctly proportioned torques are applied in the correct directions of rotation. An essential feature is that of the four members, two must react in a clockwise direction and two anti-clockwise and it follows from this that the sum of the two clockwise torques must necessarily be equal to the sum of the two anti-clockwise torques.

It now remains so to proportion the gear tooth combinations that one pair of members (e.g. the clockwise pair) will have torques of equal magnitude, in which case the two anti-clockwise members must then have unequal torques, the inequality being equal to the torque in the common shaft which connects the remaining two members together. If it is now arranged to drive the road wheels from the two clockwise members, they will thus have equal tractive torques with a balance or differential action between them, while the two anti-clockwise will then form respectively the drive input and the fixed reactance member which is fundamentally necessary if a multiplication of torque is to be obtained. Incidentally, for the input either the member with the higher or the lower torque can be chosen, and in practice it is quite convenient so to design these members that they are mounted on detachable and interchangeable flanges. Thus a choice of two ratios is available by simply unbolting the members and interchanging their positions. In conjunction with a bevel or worm drive primary reduction, this inter-changeability would provide an axle with an extremely wide range of ratios.

The accompanying illustration shows a particular example in diagrammatic form, and it might be helpful, as it is with most epicyclic problems, to ignore any consideration of how many revolutions one member makes in relation to the others and to assume instead that the whole system is static, when the relative torque values (and subsequently their speeds which are torque reciprocals) can easily be arrived at, particularly if it is remembered that a simple epicyclic train is virtually a balance beam, in which case a pivotal linear force in a given direction opposes and is equal to the sum of two balanced forces in the opposite direction ; in the other case a single torque (the planet carrier) similarly reacts against two balanced torques acting in the opposite direction of rotation.

Annulus A tends to rotate forward

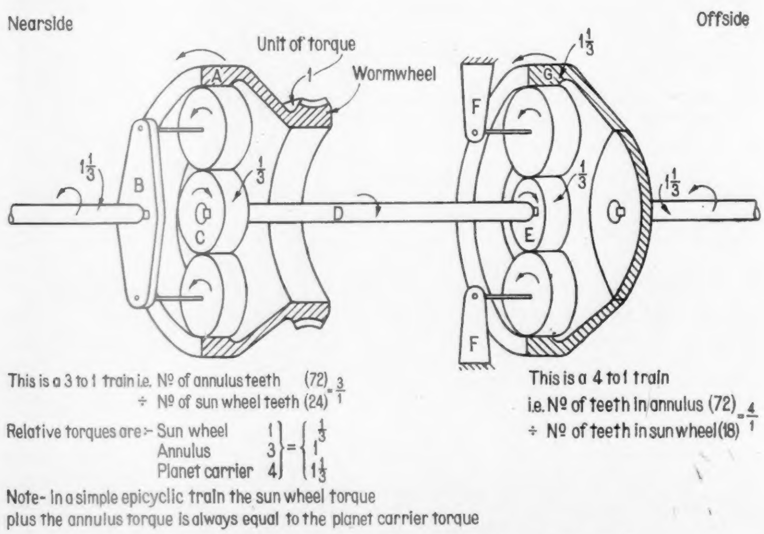


Diagram illustrating principle of Kirkstall double reduction axle.

solidly with the worm wheel and with an assumed torque of one unit. The reactive torque in the other members B and C must then be :—

(1) A *forward* torque of $\frac{4}{3}=1\frac{1}{3}$ in carrier B which is directly coupled to the drive shaft in N.S. road wheel.

(2) A backward torque of $\frac{1}{3}$ in sunwheel C and shaft D.

The drive for the O.S. road wheel is taken from shaft D, but the direction of rotation is wrong and the torque is too small. To correct this, the drive is taken through the right hand reduction train, the planet carrier F of which is fixed, and thus the *backward* torque of $\frac{1}{3}$ in the sunwheel E becomes a *forward* torque of $1\frac{1}{3}$ (i.e. $\frac{1}{3}$ times $\frac{4}{1}$ in the annulus G, output shaft and O.S. road wheel).

The two road wheels thus tend to revolve in a forward direction, and have equal torques of $1\frac{1}{3}$ units each, giving a total output torque of $2\frac{2}{3}$; this has been obtained from an input torque of 1 so the total reduction ratio must be $2\frac{2}{3}$ to 1.

It should be clear that the above torques hold good whether both road wheels are stationary, revolving at the same speed, or if one of them is stationary and the other revolving, or any intermediate condition, but if a further check of the differential action is required it can be assumed that the input annulus A is held stationary, when it will be found by checking through the gearing that one forward revolution of either drive shaft causes one backward revolution of the opposite shaft, as in the case of an orthodox differential.

CORRESPONDENCE

Correspondence on subjects of technical interest is invited. The name and address of the writer must be given, though not necessarily for publication. No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter. If a reply by post be desired, a stamped addressed envelope should be enclosed.

FRONT SUSPENSION

SIR,—I would like the opportunity of amplifying the remarks I made in that section of the Show Report referred to by Mr. R. J. Canham of Allard Motors.

The point at issue is gyroscopic reactions on the steering gear on bad road surfaces. The "swing-axle" independent suspension used by Allard (and at one time by Derby in France) is unique among its kind in introducing exactly the same amount of steering wheel "kick" on a bad surface as the "beam" axle it replaces. Though one wheel remains unaffected the other precesses through twice the angle, making the final result the same.

Now stiff springs and powerful dampers, by making the tyre do more work and holding down the displacement and precessional velocity of the wheel, can appreciably reduce gyroscopic "kick". In examining the Allard chassis I gained the impression that it was designed for high-speed competition work and to be driven by enthusiasts who would cheerfully accept the remaining "kick" as part of the game.

But the general public, on the other hand, is rapidly becoming accustomed to independent suspensions such as Lancia, Morgan or Healey, which have practically no gyroscopic kick at all, or, more generally, to wishbone systems in which precessional effects are reduced to about 30 per cent. to 40 per cent. of those given by a beam axle, this residual force being largely cancelled by the moment, about the king-pin, of the road impact causing the wheel displacement.

I have tried the car described by Mr. Canham, which is very comfortable and handles with precision. But on a really bad road the steering wheel kick is pronounced and a badly projecting manhole, for example, *momentarily* takes the wheel out of the driver's hands.

Such behaviour is, of course, a commonplace on cars with "beam" axles. But I think that the public have been educated to consider it an undesirable feature.

"THE WRITER OF THE REPORT".

CERAMIC TIPPED TOOLS

A New Product for Machining Abrasive Materials

CONSIDERABLE experimental and development work has been carried out to produce tool materials to meet the demand for tools of a hardness value between tungsten carbide and diamond. This work has culminated in the development of B.S.A.-Sintox, a product of Lodge Plugs Ltd., Rugby, England. As this is a ceramic, sinter alumina, it is distinct from high speed steel, tungsten carbide or diamond.

Owing to certain inherent properties, B.S.A.-Sintox is particularly suitable for machining materials which, though they do not require high forces for the removal of chips, are nevertheless highly abrasive and consequently likely to cause early tool failure. These abrasive materials include plastics, particularly those filled with wood, fibre or minerals, and substances such as graphite and asbestos. The great resistance to abrasion, which is an important property of the new material, not only gives increased tool life, but it also results in a reduction of wear on the clearance faces. Wear on the top faces of the tool is negligible during cutting operations under normal conditions. This eliminates the risk of sudden tool failure with consequential scrap components. Furthermore, those difficulties associated with "build up" do not occur when B.S.A.-Sintox is used, because the nonmetallic and chemically unreactive nature of the tool material precludes any possibility that the work material will weld on to the tool.

There are also other factors associated with the material which are important in maintaining a very high standard of cutting efficiency. For example, it has a very low coefficient of friction in comparison with conventional tool materials, and in addition, the extreme hardness is maintained at all times irrespective of temperature variations at the seat of cutting. The thermal conductivity is only one-half that of tungsten carbide. As a result, the heat generated during cutting is not absorbed by the tool tip, where it normally causes wear, but instead it acts as a softening agent on the swarf. It will be realised that when the swarf becomes plastic it acts as a secondary cutting medium, thus assisting the cutting action.

Machining recommendations

The machining conditions to give optimum results on a specific application will necessarily depend upon the type of material to be machined, but

certain general recommendations may be made. For example, peripheral turning and boring speeds should be as high as possible consistent with satisfactory operation of the machine. Because of the inherent properties of the material, speed does not have any detrimental effect upon the tool, and the cutting speed can be at least twice that used with tungsten carbide tools. Another important point is that the great abrasion resistance makes it possible to use a relatively large radius on a B.S.A.-Sintox tool. This allows a higher rate of feed per revolution of the workpiece for a given standard of surface finish.

In general, cutting angles that control the chip formation, that is, the rake angles, should be slightly less than those employed for conventional tool materials. Clearance angles of about 6 deg., with a secondary clearance of about 8 deg., are suitable for plastic materials. The depth of cut employed with these ceramic tipped tools may be as great as possible consistent with satisfactory chip removal, but it is also possible to use very shallow cuts for fine finishing. It is preferable to use these tips without any cutting fluid, but if a coolant is necessary, a copious flood should be directed on to the work and tool.

Tool grinding

Conventional grinding wheels are not suitable for grinding B.S.A.-Sintox tools. It is recommended that diamond wheels be used. They should be mounted on a machine that is free from vibration and during the grinding operation there should be a copious supply of "Honilo" cutting oil manufactured by W. B. Dick and Co. Bakelite bonded diamond wheels of 200-400 grit are suitable for finish grinding. For stock removal, coarser wheels up to 200 grit may be used. Although the grinding technique for these ceramic tools is the same as that used for tungsten carbide, the material can be removed at a much higher speed.

Although it must be emphasised that only diamond wheels will give a quick effortless grind, if they are not available it is possible to use green grit wheels. Those manufactured by the Universal Grinding Wheel Co. Ltd. have been found satisfactory. Grade C.80 H.V. is suitable for roughing, while C.120 H.V. may be used for finish grinding. It must be pointed out that because the rate of stock removal is considerably lower when grinding with silicon carbide wheels instead of diamond wheels, it is essential to use only a light pressure. Failure to observe this will lead to the generation of excessive heat with a consequent danger that the tip will be cracked and broken.

A range of standard turning and boring tools is available, each consisting of a tip metallised on to a high tensile steel shank. Special shapes to suit specific requirements, and in fact almost any type of tool made in hard metal can be manufactured upon request. B.S.A.-Sintox tipped milling cutters have been produced for machining graphite. Tests have shown that they are capable of giving a life between grinds at least two and a half times that of tungsten carbide or high speed steel tools. These ceramic tools may be obtained from Burton, Griffiths Ltd., Small Tools Division, Montgomery Street, Birmingham, 11.

"BRITISH ENGINEERING" FOR HOME READERS

BRITISH ENGINEERING, the sale of which was previously restricted to overseas circulation, is now available in this country. Published on the first Wednesday in each month at 2/6d., annual subscription 35/-, copies can now be ordered through a newsagent or direct from Dorset House, Stamford Street, London, S.E.1.

For many years, engineers and other technical authorities in this country have asked to see this leading technical sales engineering journal. Now they have an opportunity of doing so.

In spite of this new home distribution, the policy of confining the journal exclusively to the products and processes of the British engineering industry will not be changed. *British Engineering* covers every phase of the industry at home. Served by a panel of experts, it is backed by the full resources of Associated Iliffe Press, ensuring a particularly high standard of production. Functional colour is used to show the flow of an industrial process or the intricacies of a sectional diagram, and a regular monthly feature deals with new equipment. Each issue carries an average of 153 pages, with eight pages of functional diagrams in 3-colour and sixteen pages in 2-colour.

British Standards

THE following British Standards of interest to automobile engineers have recently been issued :—

B.S.1639 : 1950.—Simple bend test.

B.S.1620 : 1950.—Dimensions of screened magnetos (G, K and M types).

B.S.18 : 1950.—Tensile testing of metals.

B.S.1620 : 1950 deals with dimensions of base-mounted and spigot-mounted screened magnetos for internal combustion engines. It also includes dimensions for keyways and tapers and all necessary tolerances that affect interchangeability.

B.S.18 : 1950 is fundamentally the same as the standard first published in 1904. The definition of proof stress now adopted is, however, quite different from that contained in previous issues, since it has been amended to bring it in line with the practice current in industry of obtaining proof stress under load. The definition is amplified to indicate methods that may be used to ascertain if the material is satisfactory when the actual value of the proof stress is not required. (1916)

INDEXES AND BINDING CASES

Indexes and binding cases for the January-December, 1950, volume of "The Automobile Engineer", are now ready. The price of the index is 7½d. while the binding case together with index is 5s. 6d., both post free. Alternatively, readers' copies can be bound in publishers' binding cases at an inclusive cost of 12s. 6d., plus 9d. postage for the return of the completed volume. Readers should note, however, that under present conditions deliveries may be delayed.

Remittance with order should be sent to the Publishing Department, Dorset House, Stamford Street, London, S.E.1.

Pfauter Gear Hobbing Machines

IN the description of the Pfauter RS1 gear hobber, published in our February issue, it was stated that the hob saddle feed is operated by an hydraulic control unit. Actually, the hob saddle feed is controlled, not operated, by an hydraulic unit, the error having arisen in the process of translation. The address of Vaughan Associates Ltd., who handle the Pfauter products, is 4, Queen Street, Curzon Street, London, W.1.

Henry Wiggin & Company Limited, Wiggin Street, Birmingham, 16, have recently issued a leaflet dealing briefly with the Nimonic series of nickel-chromium base alloys. Since these alloys are becoming available in a variety of forms, they have been re-classified, and this leaflet summarises the available types.

CRANKSHAFT RECONDITIONING

A New Centreless Grinder for Journals and Crankpins

AN INTERESTING machine for reconditioning both the main journals and the crankpins of engine crankshafts has recently been developed by Cuthbert Machine Tools Ltd., Millmead Works, High Street, Guildford, Surrey. It is referred to as a centreless grinder, and is one in fact, since the shaft is not supported in centres. However the machine differs from all other centreless grinders in that as it does not incorporate a control wheel for imparting drive and controlling size.

The new machine, which is illustrated in Fig. 1, is basically a development of the Seest centreless crankshaft grinder for which Cuthbert Machine Tools Limited were the sole British agents before the war. During the war importation of Seest machines was impossible and the Cuthbert organisation produced a modified version that employed a 21in. diameter grinding wheel in place of the small cup wheel previously used. This modification greatly increased the rate of stock removal. After the war it was found difficult to import Seest machines and arrangements were made for a modified machine to be manufactured in this country. Actually, not only was it thought advisable to incorporate the larger diameter grinding wheel in the design, but in addition a great many other improvements have been made. In short, the machine is a completely British design developed by Cuthbert Machine Tools Ltd., specifically for crankshaft reconditioning.

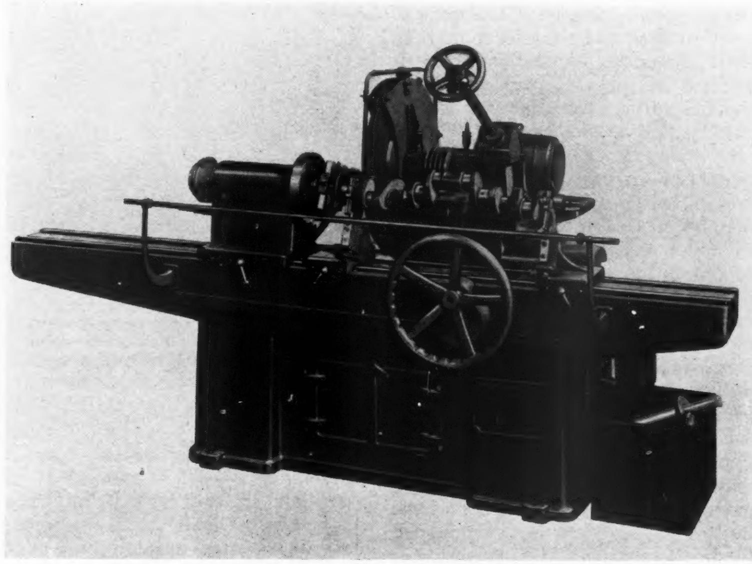


Fig. 1. Cuthbert centreless grinder set up for grinding main journals.

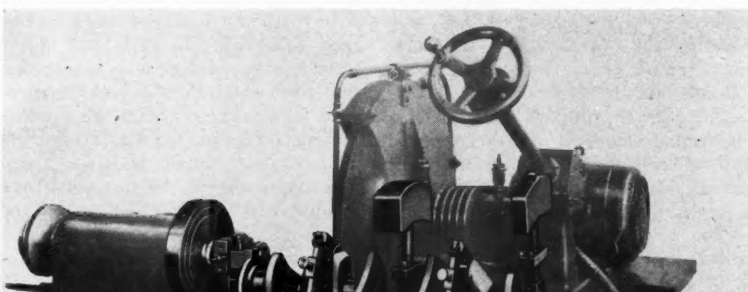


Fig. 2. Set-up for grinding No. 5 crankpin.

Design Details

There are two specially interesting features in the design of the machine. They are : (1) the method of mounting the crankshaft for grinding, and (2) the method of locating and rotating the crankshaft. For grinding all journals, the crankshaft is supported in two steadies in which the appropriate journals lie. These are so designed as to ensure that within close limits the centre of the journals being ground is always in the same position relative to the bed of the machine.

Each steady, see Fig. 3, comprises a main movable blade carrying two jaws in which the journal lies. These jaws are tipped with tungsten-carbide. The angle at which the main blade traverses is the bisector of the angle between the jaws. An adjustable finger forms a third element of the steady. It serves to retain the periphery of the journal against the two fixed jaws. Rise and fall of the fixed blade to bring the steady jaws into the position to suit a specific shaft, are controlled by a leadscrew with handwheel actuation. A scale on the main blade and a graduated dial on the leadscrew handwheel are read in conjunction. They

allow the jaws of the steady to be readily brought to the correct position for grinding a parallel journal of any given size. Grinding can be carried out opposite either of the pair of steadies. This greatly reduces the amount of handling required to complete the grinding of a crankshaft.

The workhead has been specially designed to combine the functions of work rotation and work location in an axial direction. Axial location is effected in such a manner that the steadies are free from any side load, a most desirable condition, and it also allows the steadies to be so located under the journals as to avoid any oil holes that may be present. The method of work rotation is such as to minimise any tendency for the shaft to rise out of any steady.

The drive head comprises a casting carrying a combined headstock and motor platform. Drive is taken from a gear-head motor to a three-step cone pulley to give three speeds of work rotation, which have been carefully calculated to suit all types of shafts. Each pulley is loose on its shaft and may be engaged by means of a sprung pin in the same manner as on a lathe headstock. The drive head is designed to slide freely along the bedways. It can be locked in any desired position along the bed.

At the end of the spindle carrying the driven pulleys there is a universal joint that is connected to a second universal joint through an extensible sleeve. This second universal joint is connected to a specially designed thrust head comprised of a revolving thrust plate, retained between two fixed thrust plates. The revolving thrust plate is free to move in any direction between the two fixed plates. These two plates are supported by a gimbal that permits a slight amount of angular movement of the whole drive while still preventing any axial movement of the revolving thrust plate. At the work end of the shaft carrying the revolving thrust plate there are two vee jaws that carry a slide upon which are mounted a pair of reversible chuck jaws suitable for gripping either end of a crankshaft.

With this type of centreless grinding, the attitude of the crankshaft on the machine alters as the stock is ground away. Therefore the drive mechanism must be completely flexible to allow the necessary change of attitude. All these various and essential functions are effectively performed by the Cuthbert design drive head. The purpose of the sliding jaws is, of course, to allow the necessary adjustment to be made for offset for grinding the throw pins.

A 24in. grinding wheel is employed.

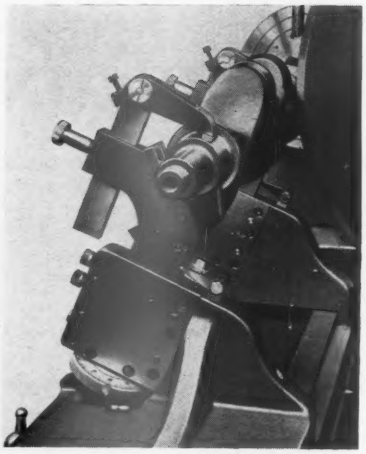


Fig. 3. A supporting steady with scales for easy and accurate adjustment.

It is rigidly mounted on a spindle running in plain bearings lubricated by drip feed. The drive to the spindle is taken from a 3 h.p. motor through four vee belts. The spindle speed is approximately 875 r.p.m. and the peripheral speed of the grinding wheel approximately 5,500 feet per minute. In-feed of the grinding wheel is effected through a handwheel mounted on a pedestal on the grinding head platform. To facilitate accurate operation, the dial on the in-feed handwheel is graduated in 0.001in. divisions. Wheelhead traverse is operated through a large handwheel on the body of the machine.

Operation

There are important differences between conventional grinding and grinding on the Cuthbert machine. For example, for grinding the main journals the usual procedure on this machine is to support the two end journals in steadies and then grind both these journals before touching the intermediate journals. In grinding a journal supported on a steady, 0.00in. advance of the grinding wheel will reduce the journal diameter by 0.001in., whereas in conventional grinding a similar advance will reduce the journal diameter by 0.002in. The reason for this is that in centreless grinding the journal is resting on its own periphery and the fixed point is the back rest, and any advance of the wheel merely reduces the distance between the wheel periphery and the rest by the same amount, and since the journal fills this space the amount removed from the diameter equals the advance of the grinding wheel. When grinding is carried out on centres, the centre of the work is the fixed point and a 0.001in. advance of the grinding wheel removes 0.001in. of stock from the radius and 0.002in. from the diameter of the journal.

This argument assumes that grinding is taking place opposite the rest. If however, grinding were started not opposite the rest, the centre of the journal would not be affected and the result would be the same as in grinding between centres, that is, the amount removed from the diameter would be twice the in-feed of the grinding wheel. Further, if grinding starts at some position not opposite the rest, a step will be formed on the journal when the cut is traversed until it is opposite the rest. Grinding must therefore start opposite the rest. This will cause the shaft to fall away with the initial cut, and once it has done so, traversing away from the rest cannot cause change of attitude and a true journal will be produced. If, as is customary, after the end journals have been ground, the intermediates are ground while the shaft is still supported on the end journals, conventional grinding practice applies.

Crankpin grinding does not create any difficulty. Most automobile crankshafts are symmetrical, that is, they have a pair of crankpins on the same axis. To grind such shafts it is only necessary to support one of a pair of pins on each steady and these pins can then be ground in a manner similar to that used for grinding main journals, see Fig. 2. A shaft so supported is, of course, out-of-balance and the out-of-balance effect is further increased by the fact that the driving chuck jaws are themselves not on centre. To allow the shaft to be brought into balance the standard equipment supplied with the machine includes two each of small, medium and heavy balance weights. There are also certain shafts for which throw-blocks must be used when the crankpins are ground. For such cases the standard equipment includes one small and one large adjustable throw - block, one fixed throw-block for Ford V8 shafts and one fixed for Bedford shafts.

In connection with this type of machine it has been argued that as the journals placed on the steady are already worn, there will be a danger of inaccuracy due to the probability that the wear will be unsymmetrical and the centre ultimately produced by a grinding wheel will be misplaced. In practice this has been found to be not only of no commercial importance but actually in favour of centreless grinding, since experience has shown that the mathematical error to which the machine is susceptible is in most cases less than that due to the personal element in grinding on a conventional machine.

Institution of Mechanical Engineers, Automobile Division

FUEL INJECTION

Wear of Equipment and Filtration of Fuel for Compression-ignition Engines

By A. E. W. Austen, B.Sc., Ph.D.*, and B. E. Goodridge, B.Sc. (Eng.), G.I.Mech.E.†

The dependence of the wear of fuel injection equipment on abrasive-particle size has been determined using closely graded abrasive prepared by air elutriation. A mechanism of wear is proposed which accounts quantitatively for the wear observed.

The particle transmission properties of filters in common use, and of possible alternative filter materials, have been measured using the same preparations of particles. Some information on the nature and particle sizes of solids in fuels is presented. Estimates of the effect of various filters in reducing abrasive wear are made for a particular particle-size distribution of abrasive. These estimates are compared with the results of a review of service experience with current filters, gauze, cloth, and felt, in Great Britain.

The choking properties of filter materials have been compared using a waxy sludge obtained from marine Diesel fuel, and a hypothesis of the choking mechanism is proposed which makes it possible to predict choking behaviour from initial resistance and thickness, and to assess, by means of a "figure of merit", filter materials of different particle-transmission properties and thicknesses.

For some services improved filtration is necessary. Papers transmit fewer particles than cloth or felt but choke more readily, necessitating the use of a large filter area; but provided the paper and free volumes on the clean and dirty sides are suitably chosen, a paper filter appears to be practicable giving a greatly improved pump element life with adequate filter element life and reasonable bulk.

THE initial aim of the work described in this paper was to devise means of evaluating the performance of various fuel filters used, or contemplated for use, on compression-ignition engines, and of specifying adequate filters for such use. The investigation is in many respects incomplete; it was decided nevertheless to publish an account at this stage since the results are of practical value in the field covered and may also have a bearing on other applications of filters.

The reasons for using fuel filters are:—

(1) To eliminate malfunctioning or failure of the equipment due to sticking or seizure of sliding parts, or blocking of fuel passages, particularly nozzle orifices.

(2) To eliminate or reduce to acceptable proportions abrasive wear.

With filters now common in Great Britain—gauze, cloth, and felt—seizure and blockage are rare and can generally be traced to faulty maintenance; but such filters do not eliminate abrasive wear. Prevention of wear is therefore a more stringent requirement and attention has been directed almost entirely to this aspect of the problem.

The most direct test of the ability of a filter to reduce wear would be to use it with a set of fuel injection equipment with the fuel it is proposed to use, and to compare the life of the equipment with and without the filter. This would be impossibly cumbrous for frequent use, and the approach used was to determine separately the wear produced by various sizes of abrasive particles; the relative merits of various filters were then determined by measuring the particle transmission for various sizes of particles in the damaging range. A knowledge of the distribution of abrasive-particle sizes in the contaminated fuel then enabled the relative lives to be expected with various filters to be estimated.

In addition to preventing wear, a filter must be capable of filtering a large volume of contaminated fuel without choking, that is, offering such a resistance to the flow of the fuel that the necessary flow is not obtained with the available pressure drop. A means of evaluating, or at least of comparing, the choking properties of filters was therefore required and has been devised.

During the investigation it became clear, both from the results themselves and from service experience abroad, that a better filter than those in common use was necessary. The bearing of the work on the selection of materials for and design of such a filter is therefore discussed.

No exhaustive search of literature has been made, since reference to a few publications of a general character (Rosenfeld 1944, Worth 1940, Langley 1943, and Pickard 1929)‡ did not suggest that it would be fruitful.

* Chief Research Engineer, C.A.V., Ltd.
† Research Engineer, C.A.V., Ltd.

‡ An alphabetical list of references is given in Appendix III.

Notation

A	Constant in wear equation.
a	Largest value of z (the minimum particle-diameter) occurring in a distribution.
b	Initial radial clearance between plunger and barrel.
e	Thickness of wax layer on internal filter surface.
F	Function representing a particle-size distribution.
f	Fraction of particles by volume per unit size range.
K	Constant representing the choking properties of a fuel or test fluid.
K_0	Coefficient of variation of particle-diameter (total).
K_1	Coefficient of variation due to scatter of size.
K_2	Coefficient of variation due to scatter of shape.
k	Fraction of surface of filter material occupied by pores or slits.
M	"Figure of merit" of filter material or filter, allowing for 50 per cent. size, and volume of filter material.
n	Number of slits per centimetre in a filter model.
q	Fraction of particles by volume smaller than x/x'.
R	Resistance§ per unit surface area of filter material.
R'	Resistance per unit surface area of filter material during choking.
s	Slit width of filter model.
T	Fraction of contaminant transmitted by a filter.
t	Thickness of filter material.
V	Volume of fuel passed through a filter.
V_c	Volume of fuel required to choke a filter element.
KV_c	Relative choking volume.
KV_{cr}	Relative choking volume for reference material.
W	Weight of material per unit area required to choke a filter material.
W_r	Weight of material per unit area required to choke a reference filter material.
w	Weight of abrasive applied to a pump line.
x	Particle-diameter.
x'	Number mean particle-diameter.
y	Wear depth (plunger and barrel).
z	Minimum particle-diameter.

Preparation of Particles

A range of closely graded powders was required for measurements of pump wear and tests on filters. For measurement of pump wear it was necessary that the material should be as hard as any likely to occur as a contaminator in fuel; and for filter tests it was desirable that it should be refractory so that, when collected on filter paper, it could be estimated by calcining and weighing. Alumina was chosen since it meets these requirements and is readily available in a range of sizes as "Alundum", a commercial grade used for lapping. For the smallest size required —2 microns and less—polishing alumina was used.

As supplied, these materials were not of sufficiently uniform size and more uniform powders down to 50 microns were prepared by sieving. For a few wear tests the polishing alumina, initially almost all less than 3 microns, was further graded by centrifuging in water to provide a powder of less than 2 microns. Intermediate sizes were prepared by air elutriation. In this process particles are injected into an upward stream of air in a cylindrical tube. Those smaller than a certain critical size are carried upwards by the air stream, which is led to a second tube of larger diameter. Here the air velocity is lower, and the critical size smaller, so that a closely graded "cut" of particles remains in the second tube. The apparatus was based on a design described by Haultain (1937).

The size distributions of the sieved and elutriated powders were determined from

§ This resistance, which is used throughout, is the pressure in dynes per sq. cm. required to give a flow of 1 cu. cm. per sec. of liquid of unit viscosity through 1 sq. cm. of the material; the unit is cm.$^{-1}$. A material with $R = 10^6$ cm.$^{-1}$ will give a flow of 0.055 gal. per hr. per sq. cm. at a pressure of 1 lb. per sq. in. with liquid of viscosity 1 poise.

Table I. Mean Diameter and Coefficients of Variation of Powders used for Filter and Wear Tests

Mean diameter x', microns	K_1	Number counted
9.0 ("500 Alundum")	0.738	294
12.4 (elutriated)	0.214	583
17.6 (elutriated)	0.200	197
32 (elutriated)	0.251	306
44 (elutriated)	0.202	526
Ideal case	0.192	—
115 sieved	0.202	408

photomicrographs. The size of a particle was taken as the distance between parallel lines tangential to it, with the lines taken always in the same direction and the particles oriented at random in the plane of the photograph. This method of measurement was chosen since the behaviour of a particle, both in causing wear and in passing through filter apertures, was expected to depend mainly on a random diameter.

The size obtained in this way is known as Feret's statistical diameter and can be shown (Walton 1948), when averaged over a large number of particles, to be equal to the diameter of a spherical particle of the same perimeter.

In this method of measurement the particles, as prepared for examination on microscope slides, tend to settle with the minimum dimension vertical; this tendency was neglected. Subsequently, it became necessary to take it into account and measure the minimum dimension.

The behaviour of powders of the same number mean diameter may be expected to depend also on the degree of uniformity and the shape. It was therefore thought worth while to measure these quantities for the powders used. The coefficient of variation, K_0, of the particle-diameter, obtained by dividing the standard deviation by the mean diameter, is made up of K_1 due to scatter of size, and K_2 due to deviations from spherical shape. K_2 varies from zero for spherical particles to 0.484 for needle-like particles. A value of K_2 was obtained by measuring the diameter of a number of particles in four directions spaced at 45 deg. The four diameters were divided by their mean and K_2 obtained as the standard deviation from unity of all these quantities. Sixty-nine particles of 128 microns number mean diameter were measured and gave a value of K_2 of 0.187.

It was assumed that the shapes of all the powders were the same and the scatter in number mean diameter was obtained from the relation, $K_0^2 = K_1^2 + K_2^2$.

The mean diameter, degree of uniformity and number of particles counted for some of the powders are given in Table I.

The mean diameter of the powder obtained for a given air flow in the elutriator was found to be proportional to the air flow and not to the square of the air flow, as it should be for a linear relation uniform across the tube section. An elutriator working according to this with particles of spherical shape in equal numbers throughout the range would yield a product with a scatter of diameter such that $K_1 = 0.192$. This value was nearly attained.

The results for all preparations may be

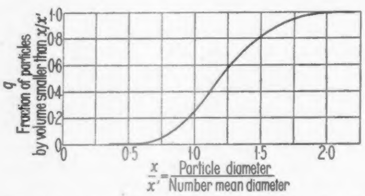


Fig. 1. Size distribution by volume of elutriated powders.

plotted together by expressing the diameter in terms of the mean diameter for each preparation. The curves were found to lie close together, and have been used to plot the single volume distribution curve of Fig. 1, in which q is the fraction by volume smaller than x/x'; x is the diameter of any particle, and x' is the number mean diameter.

The photomicrographs of Fig. 2, Plate 1, show unelutriated 500 Alundum and 12.4- and 17.6-micron powders which were prepared from it. The black particles were shown by X-ray diffraction to be also alumina with a trace of impurity.

Wear

It was decided to observe the wear caused by various sizes of abrasive particle separately in the expectation that it would then be possible to predict the wear which would be caused by any given distribution of particle sizes.

Experiments and Criteria of Wear. All wear tests were made on a conventional fuel injection system shown diagrammatically in Fig. 3.

A preliminary test was made by circulating suspensions of powders of sizes 3 microns (polishing alumina), 12½, 32, and 150 microns in fuel, separately through different lines of a six-cylinder pump. The concentration was in every case 2.3 grammes per litre (600 parts per million by volume). With all sizes, wear, as judged by visual examination, was severe in 2½ hours. This rate of wear was considered to be too different from that experienced in service for the results to be representative. Subsequent tests were made at a lower concentration, 0.03 gramme per litre (7.7 parts per million by volume). This was taken as being low enough to be in the range where wear depends on total quantity of abrasive transmitted but not on concentration.

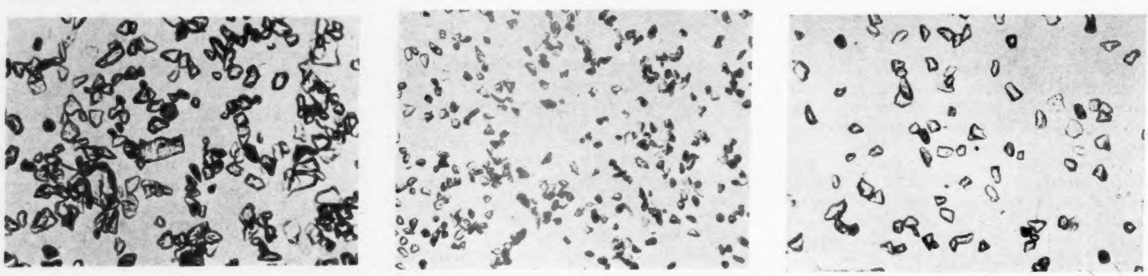
Tests were then made to distinguish between the effects of passing contaminated fuel once through the pump, and circulating a smaller volume of similarly contaminated fuel. The latter would have been more convenient. The circulated fuel caused a greater weight loss and more widely distributed wear, which was ascribed to breakage of particles on repeated passage through the system. It was decided that tests would have to be made by passing the contaminant only once through the pump.

A further test was made to explore the effect of hardness of the working surfaces on wear. No correlation between weight loss and hardness in the working range of hardness was found, and hardness measurements were not made in subsequent tests.

A series of tests was then run under similar conditions, with 32- and 12½-micron Alundum and polishing alumina of less than 2 microns. A concentration of 0.03 gramme per litre (7.7 parts per million by volume) was used throughout and the test was continued until 2 grammes had been passed through each of three lines of a six-cylinder pump. The control rod of the pump was oscillated continuously to cover a range of deliveries, and the speed was kept constant at 720 r.p.m.

Subsequently, it was doubted whether the polishing alumina was as abrasive a material as Alundum and a test was made with 3½-micron Alundum, the smallest size it was found convenient to prepare.

A number of quantitative observations were made on the worn parts, including various tests normally applied in production, weight loss measurements, and measurements of delivery. Of these none was very satisfactory as a measure of wear. There was considerable fall-off from production acceptance standards before



a b c

Fig. 2. Photomicrographs of powders used for filter and wear tests.
a "500 Alundum". *b* 12½-micron Alundum. *c* 17½-micron Alundum.
The powders shown in *b* and *c* were separated from the powder shown in *a*.

performance had deteriorated to such an extent that the part or parts would have been rejected from service. The effect of a given weight loss varied greatly with the distribution of the wear. The delivery measurements were made with the worn equipment, but with reference nozzles and with fuel at room temperature and therefore of high viscosity; variations were not as great as they might have been in service and tended to be obscured by the opposing effects of wear of the various parts. Moreover, there are no standards for rejection from service.

In general, however, the measurements did not conflict with the service experience that the factor which determines rejection from service is wear on pump plunger and barrel in the vicinity of the helix, and that the limit is set by its becoming impossible to achieve a satisfactory balance in delivery from line to line, rather than by fall-off in delivery. In service it is also found that the condition of the helix is representative of the condition of the equipment generally, severely worn elements being accompanied by severely worn valves and nozzles.

At this time the "Talyrond", an instrument with which it is possible to make roundness measurements on both plungers and bores, became available. It produces a basically circular trace on which deviations from a circular section are magnified. It was not ideal for this investigation since it does not measure absolute diameters and the gap between the plunger and barrel can only be deduced indirectly. The assessment of wear depends on the location of unworn surface which can be used as a reference; in most of these tests, however,

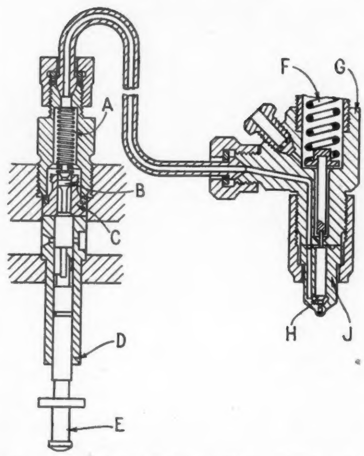


Fig. 3. Typical fuel injection pump and injector line.

some part or parts of the surface was substantially unworn. Typical traces are reproduced in Fig. 4. Such traces were used to construct the contour diagram of wear of Fig. 5.

It was decided to use such observations as a quantitative measure of wear and to take as a criterion of the end of the useful life the greatest commonly occurring wear depth found in elements rejected from service as unfit for further use. Results for the four tests already described are given in Table II, together with similar measurements on elements rejected from service. The additional experiments (tests 12, 12*a* and 13) were made to determine the dependence of wear depth on the quantity of abrasive. The wear depth varies along the length of the plungers and barrels. The values given in Table II are for the barrel just above the ports, and the plunger at the top of the helix, since wear at these points is likely to be significant as regards leakage. The wear depth averaged around the periphery (columns 5 and 6) is convenient as indicating the total amount of metal removed. Columns 7 and 8 give wear depths (for plunger and barrel) averaged on each element for the two points at which wear was greatest—at the spill port and inlet port. The wear at the latter was generally greater, and this is ascribed to displacement of the plunger towards the inlet port due to unsymmetrical fluid forces at the time wear occurred.

Mechanism of Wear. To reduce the range of measurements necessary and to permit calculation of the behaviour of abrasives of mixed size, it was desirable to formulate a quantitative account of the wear mechanism. The hypothesis advanced here cannot be regarded as fully established, as it is based on evidence which is too scanty. Nevertheless, since the hypothesis does account plausibly for the wear observed, it is the best available means of comparing the wear to be expected in service with and without filters.

Fig. 5 shows the developed surfaces of a plunger and barrel (test 12) through which 0.5 gramme of 12½-micron Alundum had passed. On each component the wear is roughly confined to two longitudinal strips and consists of straight axial scratches of uniform section for considerable lengths, often spanning the whole of the worn part. The wear patterns for other sizes of Alundum and for other quantities of 12½-micron Alundum are generally similar to the one shown; the wear depths, however, are different, and with smaller Alundum the wear is more widespread.

The wear pattern suggests that wear is caused by abrasive particles becoming trapped between the edge of the plunger and the barrel as the plunger moves across the ports on the pumping stroke. It is supposed that this trapping occurs when the width of gap through which fuel is being spilled lies between the particle diameter and zero. The particle is either scraped up the barrel by the advancing edge of the plunger or scraped down the plunger by the (relatively) descending edge of the port, depending upon which side of the port edge it happens to be.

The following assumptions are made:—

(*a*) Scratch depth is independent of particle size.

(*b*) Wear depends on the total quantity of abrasive but not on the concentration over the range used. The number of particles trapped per stroke may be estimated as indicated below. For 12½-micron Alundum the value is 3 and even for 3¼-micron Alundum the value is only 50. This is well below the number necessary to form a continuous line of particles around the periphery of the worn section, and it is therefore reasonable to suppose that the particles act independently.

(*c*) Particles of any given size cease to scratch when the total radial clearance, that is, the initial clearance plus the wear depth of the plunger and barrel, is equal to the particle diameter.

It follows that the number of particles trapped per stroke, and hence the number of scratches per stroke and rate of increase

Table II. Wear Tests and Results

Test number	Wear material		Line number	Wear depth on plunger and barrel from Talyrond trace, microns			
	Size microns	Material per line, grammes		Mean wear depth around periphery		Maximum depth	
				Individual	Average	Individual	Average
1	2	3	4	5	6	7	8
6	32	2	3		0.4		2
8	<2*	2	2		0.2		0.5
9	12½	2	2		3.5		9.0
11	3¼†	2	2		0.9		2.5
12	12½	0.5	1	2.0		6.75	
			2	2.0	1.8	6.75	6.75
			3	1.5		6.75	
12*a*	12½	1	1	2.3		8.0	
			2	2.8	2.6	8.25	8.0
			3	2.8		7.75	
13	12½	0.2	1	0.5		1.75	
			2	0.5	0.5	2.0	1.9
			3	0.5		1.75	
Service reject	..	..	1	0.6		1.5	
			2	2.0		7.5	
			3	0.8	1.3	4.0	5.3
			4	1.4		6.0	
			5	1.4		6.5	
			6	1.3		6.5	

*This test was carried out with polishing alumina; Alundum was used for the remainder.

†The pump parts used in this test had previously been run with 4 grammes per line of <2-micron polishing alumina. The helix wear, however, was not substantially greater than in test 8 and has been neglected.

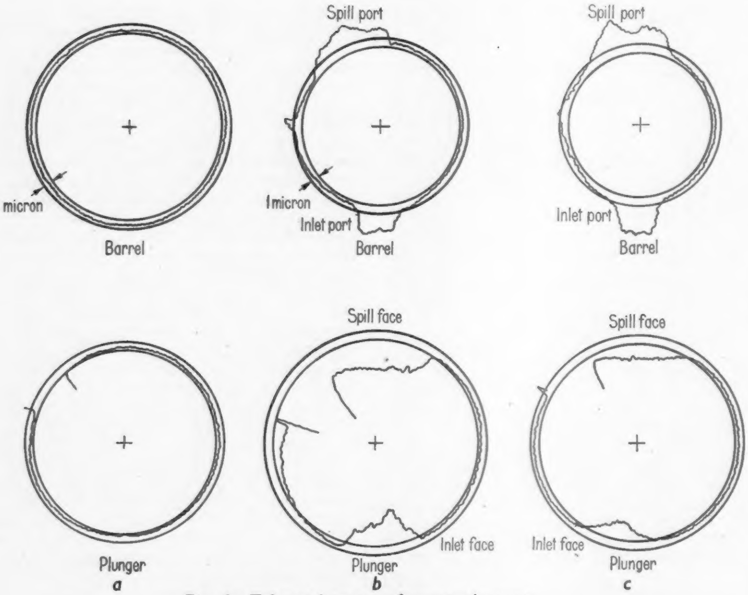


Fig. 4. Talyrond traces of pump elements.
a New element. *b* Element 1, test 12, 0.5 gramme of 12½-micron Alundum.
c Element rejected from service.

of wear depth, varies as the concentration by number, that is, for a given concentration by volume, as $1/x^3$, x being the particle diameter. The volume of fuel from which particles are strained off by the spill gap is proportional to the critical width, that is, to x. Overall, therefore, the dependence of wear rate on particle size will be as $1/x^2$ for a given concentration by volume. Thus, the initial wear rate will increase rapidly with decrease of particle size but the final extent of the wear will not exceed the particle size.

First attempts to apply this hypothesis to the wear results showed that the experimental relation between wear depth and quantity of abrasive was of the right form, but that the extent of the wear depth, for example with 12½-micron Alundum, reached too low a limiting value. This conforms with the findings of Thayer (1944), who observed the wear on a high pressure hydraulic system; it was found that the increase of clearance was less than the particle size in this range.

It was suspected that this was due to the abrasive particles having smaller dimensions in the vertical direction that in the horizontal directions in which they had been measured. This was confirmed by microscopic examination of 116-micron Alundum. This was found to settle on the microscope slide with the minimum dimension vertical, and the mean height was half the mean projected diameter. The assumptions are therefore modified as follows:—

(*d*) Particles of a given size cease to scratch when the minimum particle dimension is equal to the radial clearance.

(*e*) The shape of the particles is independent of particle size. Qualitative microscopical examination supports this.

The relative wear to be expected from different amounts of different sizes of abrasive can be calculated as shown in Appendix III. The results are shown in Fig. 6 for the Alundum preparations of nominal size 3½, 12½, and 32 microns and an initial radial clearance of 1 micron. The one disposable constant in the expression deduced depends upon the area of cross-section of a single scratch. In Fig. 6 it has been chosen to give the best fit for one 12½-micron point. The good fit obtained for the other 12½-micron points and for the 3½- and 32-micron points is an indication of the extent to which the hypothesis fits the experiments.

A further check may be obtained from optical interference observations of scratch depth made as described in Appendix II. In Fig. 7, Plate 2, interference photomicrographs for three parts of a plunger worn with 12½-micron material are reproduced. The pattern inside the worn patch is confused and the best estimate of the individual scratch depth can be obtained from Fig. 7*c* from the relatively unworn part. The vertical scratches occurred during the wear test. To obtain the order of magnitude of scratch cross-section, a single scratch has been taken to be of triangular cross-section, 2 microns wide at the base by ⅓ micron deep.

From the area of cross-section of the metal removed and the area of the single scratch the number of particles causing the wear was calculated. This, together with the total number of particles used in the test, gave the fraction trapped and causing scratches. The value obtained was 1 in 90.

According to the wear hypothesis, the fraction of the total number of particles trapped is given by the ratio of the volume of fuel passing through the port while the plunger moves through the critical gap, in this case through 12½ microns, to the total quantity of fuel delivered. This gave a value for the fraction of particles trapped as 1 in 130 which is in good agreement with the value calculated above from the observed wear.

The additional curve in Fig. 6 for zero to 30 microns rectangular distribution of Alundum was calculated in the same way as the other curves.

Filters

Particle Transmission. Although it was recognized that the particle-transmission properties of a filter might change in service, it was nevertheless decided to measure only the initial properties, since no information on the particle-size distribution in service was available and it was known that very small quantities of abrasive could cause serious wear. All tests were therefore made with a quantity of contaminant less than that required to form a single layer on the area of the filter. Subsequently, this procedure was justified since it was found that, with wax choking, the particle-transmission properties of sheet materials changed little up to the point at which the filter was almost fully choked.

Tests were made by supplying to the test filter fuel contaminated with the appropriate quantity of elutriated Alundum powder as used for the wear tests. The quantity transmitted was estimated by passing the outflow through chemical filter paper fine enough to retain all the test material. The filter paper was then calcined and the quantity of Alundum determined by weighing. It was found that if the contaminated fuel were followed by successive quantities of clean fuel more of the contaminant was washed through. This is illustrated in Fig. 8, in which the fraction of the contaminant, transmitted, T, is plotted against $1/\sqrt{V}$ (V is the volume of fuel) for two sizes of Alundum, and a cotton cloth filter. In service there would be ample opportunity for this washing through to proceed to finality and the fraction finally transmitted is therefore the quantity which should be measured. In this work this has been done by using plots such as those of Fig. 8, the fraction finally transmitted being obtained by extrapolating to $1/\sqrt{V} = 0$.

Some tests were made to explore the effect of pulsating flow from a feed pump compared with the steady gravity flow first used, and of vibration by clamping the filter to an engine. The effects were obscured by variability of the results, but both pulsating flow and vibration tended to give greater transmission. The effect of pulsating flow appeared to be greater than that due to vibration but did not increase the fraction transmitted by more than 50 per cent. Since both the nature of the flow and vibration vary between installations subsequent tests on complete filters were made with approximately steady flow from a gear pump at the maximum service flow (10 gal. per hr.), and without vibration.

For tests on filter materials, 7-centimetre diameter circles clamped between rubber rings were used. To reproduce the service

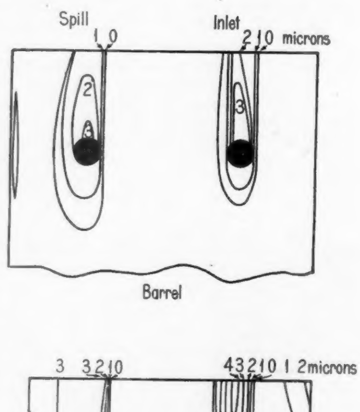


Fig. 5. Wear depth contour diagram.
Element 1, test 12, 0.5 gramme of 12½-micron Alundum.

flow per unit area would have necessitated running tests for several hours to give convenient volumes of fuel. Tests were made at a flow per unit area of about five times the service value, using a vacuum pump.

The results, more particularly on complete filters, were rather variable. Some of the variation was found to be due to test material sedimenting in the bottom of the filter pot. Attempts to prevent sedimentation failed, but it was found that, for the closely graded powders, the particle-size distribution of the sedimenting material was indistinguishable from that of the material applied. The effect of sedimentation was therefore eliminated by weighing the sediment and subtracting it from the quantity nominally applied. This procedure reduced the experimental scatter and gave reasonable agreement for cotton cloth between tests on complete filters and on small circles.

Results for a number of filters used or contemplated and for some possible alternative materials are given in Table III.

Possible sources of variation not fully explored in these tests are :—

(1) Variations in the test material from piece to piece.

(2) The manner in which the initial application of contaminated fuel is made.

(3) Agglomeration of the test powder.

(4) Uncertainty in the extrapolation made to deduce the quantity finally transmitted. For example, despite the linear plot of T against $1/\sqrt{V}$ generally obtained, it is possible that transmission depends rather on the number of interruptions of the flow.

(5) Dependence of transmission on flow rate ; it is unlikely that this effect is large apart from sedimentation, in view of the fair agreement for cotton cloth between tests on complete filters and on small circles at five times the flow per unit area.

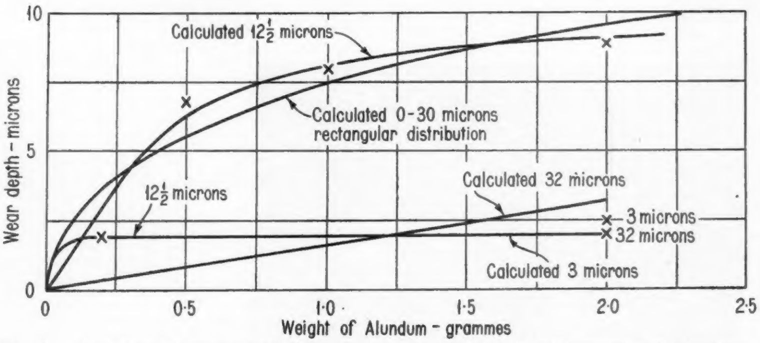


Fig. 6. Variation of wear depth (pump plunger plus barrel) with weight of Alundum. × Experimental points.

Despite these uncertainties the results are believed to represent the relative behaviour of the various materials with sufficient accuracy for present purposes.

It is an attractive simplification to suppose that a filter has a "cut-off" size such that it retains all larger particles and transmits all smaller ones. If this be the case the cut-off size may be determined from the fraction transmitted and a cumulative size distribution curve for the powder used (Fig. 1). In these tests the apparent cut-off size was found to vary with the size of test material as shown in Table IV.

It is clear that filters cannot be described in terms of a cut-off size.

The particle-transmission properties of a filter can thus only by fully described by a curve of fraction transmitted against particle size. If the test preparations were truly of uniform size, tests with a number of them would give this information. In fact, however, there is a considerable scatter of particle size. The possibility of deducing true particle-transmission curves from the experimental results has been considered. It was shown that the true curve is considerably sharper than that plotted from the experimental results themselves, but that measurements for a greater number of mean sizes would be necessary to make it possible to deduce with any precision the true particle-transmission curve. The apparent transmission curve has therefore been used. Typical curves are shown in Fig. 9.

For the rough comparison of different filter materials and for calculations on models intended to represent the behaviour of practical materials, it is sometimes convenient to take a single value for the pore size of a filter. For this purpose the 50 per cent. size has been used. This is defined as the size of particle such that 50 per cent. by weight is transmitted by the filter and 50 per cent. retained. Where sufficient measurements have been made on a material to draw the transmission curve this quantity has been taken from it. In other cases the 50 per cent. size has been derived from the nearest available experimental point and the slope of the transmission curve of a similar material.

Filter Choking. In descriptive and advertising matter on filters it is common to state the initial resistance, generally in the form of a flow for particular fluid at a certain pressure drop. It is necessary that a filter should give the desired flow at the available pressure drop, but this in itself is not sufficient. It is also necessary that a filter should continue to give the desired flow after filtering a large volume of contaminated fluid. Ideally, a filter specified for a given service should be rated according to the volume of fluid it will filter without choking ; this is not necessarily directly related to initial resistance, which, provided it is low enough, is of no particular significance.

With fuel filters the nature and extent of contamination of the fuel and conditions of service are so variable that it is not practicable to define a volume of fuel which a filter will pass without choking. An attempt has been made to devise a means of comparing the choking properties of filters and a test has been produced which is believed to represent satisfactorily the relative behaviour in services of such a nature that choking is a serious problem.

First attempts at choking tests were made with Alundum powders as the choking material. It was found that the resistance produced by a given quantity of contaminant was extremely capricious, and was due to the formation of a "cake" on

Table III

Particle Transmission of Filters and Filter Materials

Filter material	Mean particle size, microns					Estimated 50 per cent. size, microns
	3¼	12½	17½	32	44	
Cotton cloth	99	85	73	30		25
Felt cloth				10		
*Block felt			50	2		17
3.7-ounce nylon cloth		18				9
*Multiple wire gauze					95	
*Clamped paper stack			16	6		
Sintered bronze A	16					1.8
Sintered bronze B		85				
†Paper fuel filter 1 (embossed, heavily resin impregnated)	55	35				4
†Paper fuel filter 2 (lightly resin impregnated)	24	6				1.8
‡Rag paper, craped, thick	52					
‡Rag paper, craped, thick, 1 per cent. resin impregnated	30	13				
‡Rag paper, craped, thin, 1 per cent. resin impregnated	34	13				2
"Kraft" paper, craped	37	10				2.3
Chemical filter paper Whatman No. 54	39	10				
Chemical filter paper Whatman No. 50	0	0				

The values are the percentages finally transmitted.
*Test on complete filter ; all others were tests on 7-centimetre diameter circles.
†These differences were due to the different papers and not to the different impregnations.
‡These papers were all similar apart from thickness. Sedimentation was in all cases prevented or allowed for.

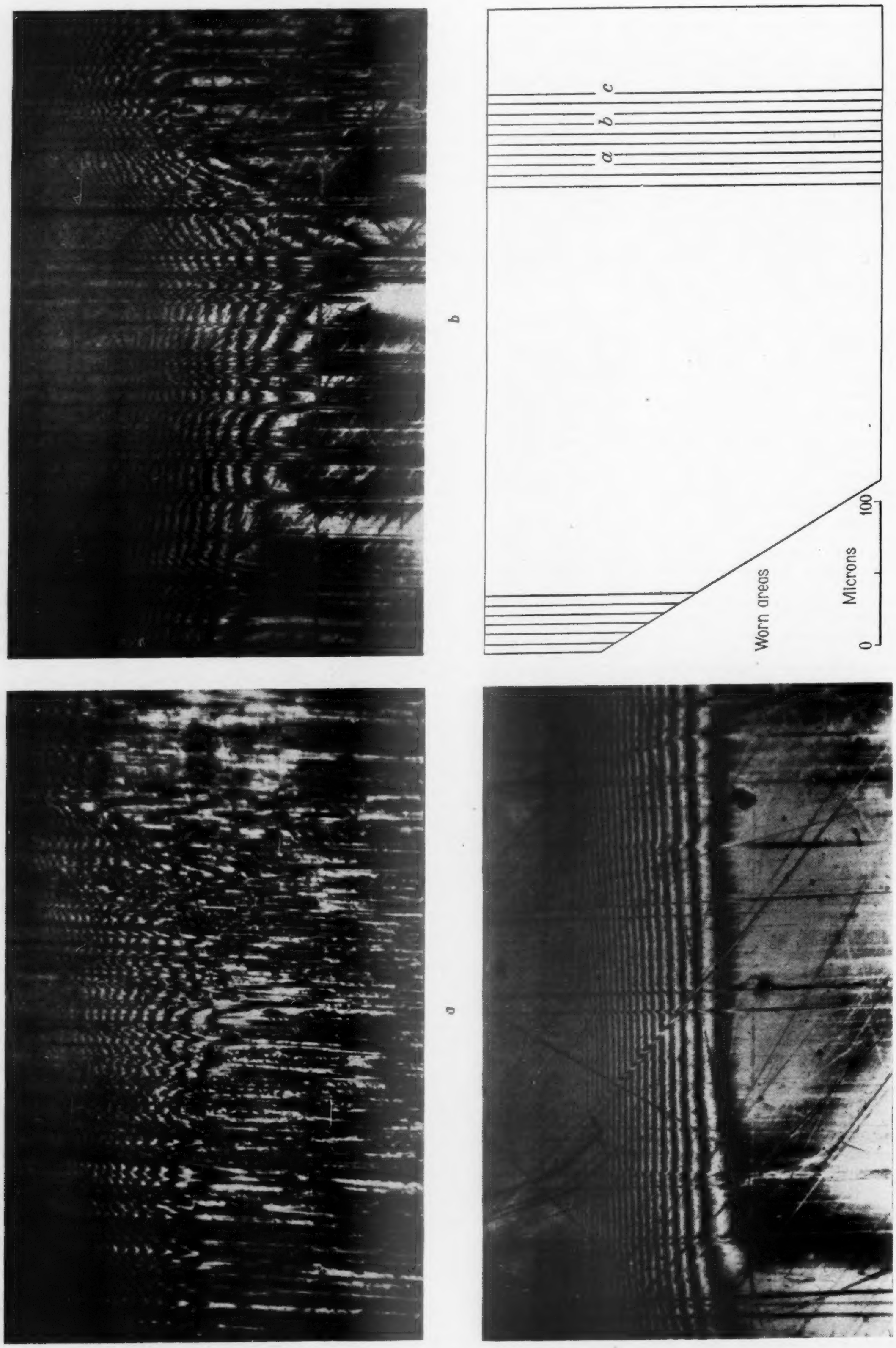


Fig. 7. Interference photomicrographs of worn plunger 12½-micron Alundum was used ; 1 fringe $\simeq$ 0.25 micron.

Table IV

Apparent Cut-off Size of Cotton Cloth

Mean size of test powder, microns	$3\frac{1}{4}$	$12\frac{1}{2}$	17	32
Apparent cut-off size, microns	6.3	19	24	33

the surface of the filter material. Also, the quantity of solids required to cause complete choking was much greaterthan was found on filters withdrawn from service. Examination of filters used in Africa showed that the choking material was soluble in organic solvents and this phenomenon proved to be well known to fuel suppliers. The material is a waxy sludge consisting of the higher petroleum fractions.

Attempts were then made to choke filters with suspensions of paraffin wax, and later with longer chain paraffins—"Winnothene" (a product intermediate between paraffin wax and polythene) and polythene. All behaved capriciously, and although the longer-chain materials caused choking with about the appropriate weight of material, the choking layer was much more bulky than in the service case.

Samples of wax sludge centrifuged from "B" grade marine Diesel fuel were then obtained and were found to cause choking very similar to that found in service. It was therefore decided to make choking tests with this material. It has the disadvantage that it is not a widely available, well defined material, but this objection was to a limited extent overcome by obtaining a large supply.

Tests at different concentrations showed that the effect of concentration was small in the range 0.8 to 2.0 grammes per litre, choking being dependent upon the total quantity applied. Although more consistent than paraffin, the petroleum sludge still sometimes gave results differing by a factor of three. The difficulty was circumvented by making a test with each suspension just before or after use on a reference filter material; the material chosen was "Whatman No. 54" filter paper.

Some typical choking curves are shown in Fig. 10. The relative weights of sludge required to raise the resistance of various materials to the critical value are shown in column 8 of Table V. These weights refer to a filter element of area 800 sq. in.

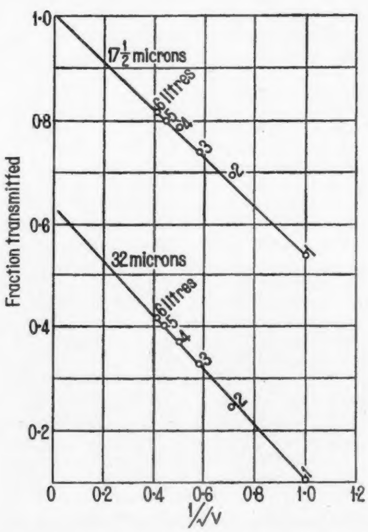


Fig. 8. Solids washed through cloth filter by successive quantities of clean fuel, for two sizes of particles.

V is the volume of fuel in units of 1 litre.

Filter Models and Criteria of Filter Materials. An attempt is made here to devise a hypothesis of the mechanism of the choking of filters by wax sludge. The objects are to relate the wax choking properties to other more easily measured quantities and to provide a basis for the comparison of materials of different particle transmission properties and thickness.

Most of the filter materials concerned consist of assemblages of fibres mainly lying perpendicular to the direction of fluid flow. The 50 per cent. sizes, at any rate of the papers, indicate that, for all or nearly all paths through the medium, the fibre spacing is, at some point in the path, small compared with the fibre diameter, for example, 2 microns as against 16 microns. The most appropriate simple model is thus an array of parallel, plane slits.

The resistance per unit surface of such an assembly is given by

$$R = (12t/s^3n) \quad .. \quad .. \quad (1)$$

Here, s is the slit width in centimetres taken as the 50 per cent. size,

$$n = k/s \quad .. \quad .. \quad (2)$$

Values of k calculated from the measured values of t, s and R are given in Table V. All values are of the appropriate order of magnitude. The fact that some values are greater than unity indicates that the model would be a closer approximation to the real material if the length of the slits in the direction of flow were taken as a fraction of the thickness of material rather then equal to the thickness; all values of k could then be made less than unity. This adjustment has not been made since no use is made of the absolute values of k.

It is now supposed that the wax builds up as a layer of uniform thickness e on both faces of each plane slit and that e is proportional to the volume of fuel passed, inversely as the slit inlet area k and inversely as the slit width s.

$$e = (KV/ks) \quad .. \quad .. \quad (3)$$

where K is a constant for the particular wax suspension.

The resistance during choking is given by

$$R' = R\left(\frac{1}{1 - \frac{2e}{s}}\right)^3 \quad .. \quad (4)$$

When this expression is compared with experimental results, it is possible, by choice of the constant K, to obtain a fit at a particular point. In Fig. 10, K has been chosen to give a fit for Whatman 54 paper at half the choked resistance for a particular application. Experimental curves and calculated curves using the same value of K for another paper, and for cloth and felt are compared in Fig. 10.

The theory represents the relative choking behaviour of different materials fairly well. The calculated curves, however, rise rather more steeply than the experimental curves and do not show the fall off in rate of rise later. Actual filter materials, when choked, show an apparently continuous layer of wax on the input side and yet continue to pass some fuel. It is thus clear that the wax deposit is to some extent permeable. This is believed to account for the different shapes of the experimental and calculated curves. To introduce a permeable choking medium into the theory would involve rather cumbersome mathematics which would not be justified at this stage.

In practice, at the choking point R'/R is large and occurs at values of $2e/s$, approaching unity where R'/R varies rapidly with e. Most filter media choke at values of $2e/s$ lying between 0.8 and 0.9 and no serious error is therefore introduced by taking the choked values of e_c as given by $(2e_c/s) = 0.85$. From this $e_c = 0.42s$.

From equation (3), $KV_c = kse_c = 0.42s^2k$, and from equations (1) and (2) $k = (12t/Rs^2)$, thus $KV_c = (5t/R)$, the 50 per cent sizes disappearing from the equation.

Column 6 of Table V gives the quantity $5t/R$ for various filter materials. Column 7 gives the expected choking volume relative to that for Whatman 54 paper, that is, KV_c/KV_{cr}, and column 8 the corresponding experimental value. Column 9 gives the ratio (experimental/calculated) = [(column 8)/(column 7)]. If the theory gave a perfect fit all these values would be unity. Although the materials include a range of choking values of 200 to 1 the greatest discrepancy between the theory and experiment is a factor of 0.5 which is not much greater than can be ascribed to the uncertainties of the experimental values. It is therefore considered that, although this hypothesis is not a complete account of the phenomena, it represents behaviour well enough to justify its use as a basis of comparison of materials.

Filter materials which transmit large particles are to be expected to choke less readily than finer filters. To compare the efficiencies of such different materials it is necessary to allow for the 50 per cent. size. According to the model the resistance of a single slit varies as $1/s^3$; but the number of slits is proportional to $1/s$. Thus the resistance per unit area varies as $1/s^2$, and to compare the choking properties allowing for the 50 per cent size KV_c/s^2 is taken.

In practical filters it is also necessary to take into account the bulk of the material, since what matters is the quantity of fuel which can be filtered per unit filter volume.

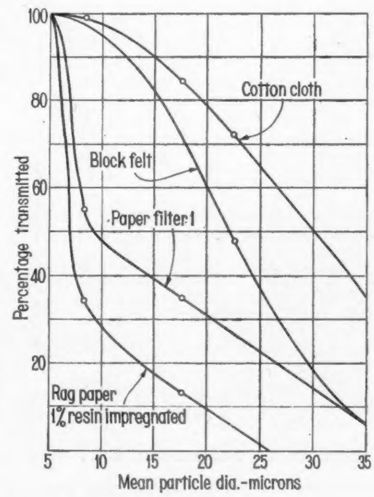


Fig. 9. Particle transmission of filter materials.

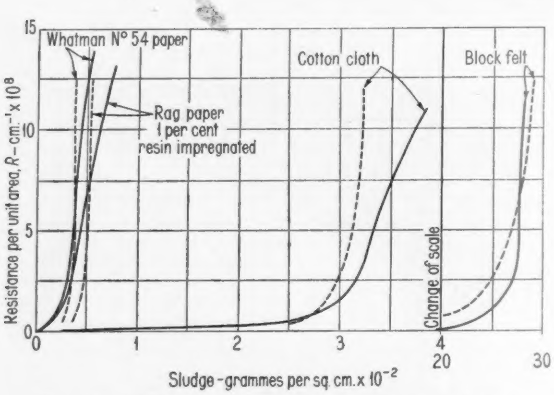


Fig. 10. Choking of filter materials by waxy sludge.
– – – – Calculated.
———— Experimental.

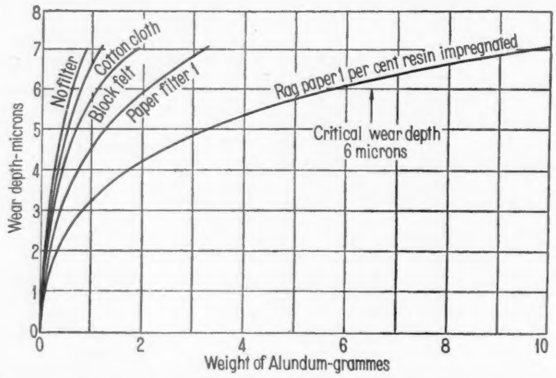


Fig. 11. Calculated wear caused by zero- to 30-micron rectangular distribution of Alundum with various filters.

The appropriate overall figure of merit thus becomes $M = (KV_c/s^2t) = (5/Rs^2)$. Values of this quantity for the materials tested are given in Table V.

Solids in Fuels

A knowledge of the particle-size distribution of abrasive solids in fuel is necessary to compare the effectiveness of various filters. If the quantity and wearing properties relative to Alundum were known as well it would be possible to predict service lives. When a filter is designed it is necessary to provide sufficient free volume to accommodate the quantity of solids expected in the servicing period; the total solid content is the appropriate quantity for this purpose.

The data available initially were very scanty and a systematic investigation has not been made. Some measurements had, however, been made, and the authors had some observations arising from particular cases of service complaints of excessive wear.

Total solids varied from 3 parts per million by weight for a storage tank in Great Britain to 230 parts per million for a vehicle tank in Africa. Generally, more than half the total solids were organic, and of the rest most consisted of iron compounds while rather less than half was estimated as silica which may have been present as silicon-containing compounds. Where a comparison was made between a vehicle tank and the corresponding fuel supply, the solids content of the vehicle tank was generally greater; it was concluded that corrosion of the vehicle tank was responsible, although this did not always occur.

If, where silica was estimated, it were assumed that all of it would pass through the filter and that the material was as damaging as Alundum, the life of the injection equipment should have been shorter (about one-quarter) than that to be expected in such service. In other cases an attempt was made to estimate, by microscopic examination, the fraction of crystalline material and it was verified that some of the material was capable of scratching hard steel. Here again, if only the crystalline material were taken as being as damaging as Alundum the lives estimated were much too short. It has not been possible to identify or estimate the damaging material, or to distinguish between the possibilities :—

(*a*) that most or much of the service contaminant causes wear but that it is much less damaging than Alundum, and

(*b*) that only a small fraction is capable of causing appreciable wear.

A determination of particle-size distribution of solids in a vehicle tank showed roughly a rectangular distribution by weight from zero to 30 microns (per cent. per micron constant). This was supported by qualitative observations in other cases, though some larger particles, up to 70 microns, were occasionally seen. The zero-to 30-micron distribution has been assumed for comparisons of the effects of filters on wear. For filter design purposes 50 parts per million by weight has been taken as the greatest concentration likely to be of frequent occurrence. This is roughly equivalent to a superficial volume of 33 per million volumes of fuel, or 150 cu. cm. per 1,000 gal.

Practical Application

The wear to be expected without a filter and with certain filters is considered here, and is compared with available information on the service behaviour of present filters. The effects of filter area and the bearing of this work on filter element design are discussed.

Wear without a Filter and with Certain Filters. It is assumed that the particle-size distribution of the abrasive constituent of

Table V. Experimental Data, Model Parameters and Figures of Merit for Various Filters

Material	Thickness of filter material t, cm.	50 per cent particle size, microns	Resistance per unit area R, 10^6 cm.$^{-1}$	$k = \frac{12t}{Rs^2}$	KV_c, 10^{-8} cm.^{2}	$\frac{KV_c}{KV_{cr}}$	$\frac{W}{W_r}$	Ratio $\frac{W}{W_c} \Big/ \frac{KV_c}{KV_{cr}}$	Figure of merit M, 10^{-2} cm.$^{-1}$
1	2	3	4	5	6	7	8	9	10
Cotton cloth	0.036	25	1.05	0.066	17.1	7.4	8.7	1.17	76
Block felt	1.8	17	4.9	1.5	175	76	68	0.9	35
Paper fuel filter 1	0.04	4	2.7	1.1	7.4	3.2	1.6	0.5	1,140
Paper fuel filter 2	0.073	1.8	5.8	4.6	6.3	2.7	1.4	0.52	2,680
Rag paper, thin, 1 per cent. resin	0.022	2	3.4	1.9	3.24	1.4	1.3	0.93	3,680
Rag paper, thick, 1 per cent. resin	0.032	2	5.9	2.6	2.7	1.17	1.1	0.94	2,100
Rag paper, thick, untreated	0.032	2	5.0	1.9	3.2	1.4	1.4	1.0	2,520
"Kraft" paper	0.023	2.3	9.8	0.53	1.2	0.52	0.3	0.58	1,000
Chemical filter paper Whatman No. 54	0.019	2.5	4.1	0.9	2.3	1	1	1	1,940

Columns 2, 3, 4 and 8 are measured values.
Column 5, values of k, the fraction of the surface occupied by apertures should not exceed unity. The large values calculated here are discussed in the paper.
Column 10, figure of merit, choking volume weighted to allow for particle transmission and bulk of filter material.

service contaminator is the same as for the one available determination on total solids, that is, a rectangular distribution of from zero to 30 microns; that the relation between the size determined microscopically and the minimum particle dimension is the same as for Alundum, and that the relative effects of different sizes and quantities of the service contaminant are the same as for the same sizes and proportional quantities of Alundum. The service concentration is not known and lives must therefore be expressed in terms of life with no filter; these relative lives will be the same as for a zero- to 30-micron distribution of Alundum.

The relation between the weight of zero- to 30-micron Alundum and wear depth has been calculated as outlined in Appendix III, using the data already given. The result is plotted in Fig. 6 and reproduced in Fig. 11. The wear to be expected with various filters has been calculated by using the filter transmission curves to obtain the quantity of the zero- to 30-micron distribution transmitted in each 2-micron size range. The rest of the calculations then follow as before. Results for cotton cloth, block felt, paper fuel filter 1, and the thin rag paper are plotted in Fig. 11. The corresponding relative lives, taking a maximum tolerable wear depth of 6 microns, are given in Table VI. These estimates of life make no allowance for any abrasive which might be removed in the filter by sedimentation. In the filter tests at the maximum service flow the amount removed by sedimentation was not large. It is possible that in some services the effect of sedimentation would be considerable.

Service Performance of Present Filters. The information available consists of a review of fuel pump element life containing returns made in 1944 by fifty-nine operators in Great Britain.

For most of the operators, the type of service was classified as public service vehicles or haulage, and filters as cotton cloth, block felt, or wire gauze. There is a significant difference of life between public service vehicles and haulage; the means are 294,000 and 128,000 miles respectively. This difference is commonly ascribed to the use by public service operators of their own fuel storage tanks in which time is allowed for cleaning by sedimentation, with the result that public service vehicles, on the average, run on cleaner fuel. It could, however, be due to the different conditions of operation, more frequent stops and periods of slow running on public service vehicles giving more opportunity for sedimentation in filter pots. In both groups the differences between filters were small and not significant, despite the fact that the gauze filter stops only particles greater than 100 microns.

The service data support the findings of the previous section that the effect of filter material in present filters is not large. No information is available on the life with no filter and the importance of sedimentation in the filter pot cannot therefore be estimated.

While no body of evidence comparable with that discussed above for normal service in Great Britain is available for service abroad, a considerable number of service complaints from abroad and a few in Great Britain have indicated that service lives much shorter than those quoted above may occur.

Filter Element Design. Service experience has shown that improved filtration is essential to achieve better injection equipment life in some services. Even when long service is already achieved it is likely that improved filtration would result in better maintenance of initial performance.

Table VI
Relative Lives of Fuel Pump Element on Zero- to 30-micron Alundum Distribution, Neglecting Sedimentation

Filter	Life
None	1
Cotton cloth	1.3
Block felt	1.8
Paper fuel filter 1	3.4
Rag paper, thin	8.5

For service reasons, attention has been confined to filters of the type in which the filter element is discarded when choked, and replaced by a new one. Choking of cloth filters is occasionally troublesome in service, and it is therefore thought that a filter should not choke more readily than the cloth filter. For a finer filter material this requires a large filter area and, for reasonable filter size and expense, restricts the choice of the materials tested to paper.

Increase of filter area has two effects:—

(*a*) since flow per unit area for a given required total flow is decreased, the resistance per unit area at the choked point is increased, and

(*b*) for a given resistance per unit area the fuel filtered is proportional to filter area.

If the resistance rises very rapidly at the choked point, as in the theoretical curves of Fig. 10, the first effect is negligible and the choking volume is proportional to the filter area. On the other hand if the resistance is proportional to the volume of fuel filtered the choking volume is proportional to the square of the filter area. Actual behaviour of papers appears to be intermediate and, in a particular case, doubling the filter area increased the choking volume by about three times.

If the large area required in a paper filter is not to result in a very bulky filter, the paper must be accommodated in a close-packed form. The construction should be such that the space on the clean side is only just great enough to permit the exit of the fuel without appreciable resistance. If only wax choking had to be considered the space on the dirty side would be similarly restricted, since wax chokes with a negligibly thin layer. In practice, however, provision must be made for the accommodation of solids as well as wax.

The ideal design procedure is therefore as follows:—

(1) A filter material should be chosen which will give the desired pump-element life relative to that with an existing filter. If more than one material is available the one with the highest figure of merit should be chosen.

(2) The filter element life (volume of fuel) required relative to that for an existing filter should be decided. This will determine the filter area required.

(3) The quantity of solids to be expected in this quantity of fuel should be determined; this will determine the free volume required on the dirty side of the filter material and hence the volume of the filter element.

This procedure has resulted in the design of a filter element which should give a greatly improved life of the injection equipment, and a filter element life slightly better than present cloth filters, with reduced filter element and pot volumes. "Pilot" production, and service tests both in Great Britain and abroad are in progress.

Acknowledgements. The early work, including the construction of the elutriator and preliminary filter tests, was done by Mr. K. Gilbert.

The authors acknowledge the help and encouragement given by many of their colleagues during the investigation, and in particular that of Mr. H. Charlton, who was responsible for making the wear tests.

Thanks are due to Mr. L. E. Lowe of Shell Petroleum Co., Ltd., for information about wax choking contributed at a critical stage, the Thornton Research Centre for their work on solids in fuels, and Taylor, Taylor and Hobson, Ltd., who made the Talyrond measurements on plungers and barrels.

The authors are indebted to C.A.V., Ltd., for permission to publish this paper.

APPENDIX I

Interference Method of Observing Scratch Depth

A replica of the scratched cylindrical surface is made by dipping the part into a preparation of methyl methylacrylate monomer (Pearson and Hopkins 1948) with the addition of 5 per cent of butyl phthalate—a plasticizer. The film is then polymerized by exposure to an ultra-violet lamp for 30 minutes and stripped off.

The replica is then placed on a piece of optically flat glass, with the scratched surface towards the glass, and with a strip of gold leaf 12 microns thick between the replica and the glass along the edge corresponding to the square end of the plunger. The replica is forced into close contact with the glass by covering with a rubber sheet having a sealing rim round its edge and evacuating the space between the rubber and glass. A wedge-shaped space is left between the replica and glass in the neighbourhood of the gold leaf strip.

This wedge is then observed under the microscope with vertical illumination using monochromatic light. A tungsten source and Ilford spectrum yellow filter were found to be little inferior to a sodium-vapour lamp and were used, being more convenient. Interference fringes are observed in the wedge. For a replica from an unworn lapped surface the fringes are straight and parallel to the gold leaf strip. Any small-scale imperfections in the surface are shown up as displacements of the fringes which may be interpreted as contour lines of the surface of the replica.

Typical photomicrographs for a plunger surface worn with 12½-micron Alundum are shown in Fig. 7, Plate 2.

APPENDIX II.

Wear due to Abrasives having a range of Particle Sizes

If the distribution of sizes is $(df/dx) = F(x)$, the distribution of the minimum particle-diameter, that is, the smallest spacing between parallel planes which include the particle is taken as $(df/dz) = 2F(2z)$.

If δw is a small increment of w which increases wear depth from y to $(y+\delta y)$, the weight of particles in δw of size between z and $(z+\delta z)$ is $\delta w \frac{df}{dz} \delta z$ and increases the wear depth by $\frac{A}{z^2} \delta w \frac{df}{dz} \delta z$.

Particles such that $z < (y+b)$ produce no wear,

hence $\delta y = A\delta w \int_{(y+b)}^{a} \left(\frac{1}{z^2}\frac{df}{dz}\right) dz,$

or $\frac{dy}{dw} = A \int_{(y+b)}^{a} \left(\frac{1}{z^2}\frac{df}{dz}\right) dz$

The quantity of the right-hand side was not generally expressible in a form convenient for integration. It was therefore integrated numerically for values of y from zero to $(a-b)$. The relation between y and w was then also computed numerically.

APPENDIX III

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INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during March :—

LONDON
Tuesday, 13th March, 5.30 p.m. General Meeting at Storey's Gate, St. James's Park, S.W.1. Paper : "The Use of Wire Resistance Strain Gauges in Automobile Engineering Research, with Particular Reference to Strain in Vehicle Structure", by J. R. Bristow, B.Sc., Ph.D., A.M.I.Mech.E. P. Metcalf, B.A., B.Sc., and C. H. G. Mills, B.Sc. (Eng.), A.K.C.

BIRMINGHAM
Monday, 12th March, 6.45 p.m. General Meeting at the Chamber of Commerce, New Street. E. N. Soar, A.M.I.Mech.E., will deliver the Thomas Hawksley Lecture entitled "The Supercharging of Internal Combustion Engines", in the absence of the author, Sir Harry Ricardo, B.A., LL.D.

DERBY
Monday, 12th March, 7.15 p.m. General Meeting in the Midland Hotel. Paper : "Engineering Aspects of the B.R.M. Car", by Raymond Mays.

LUTON
Monday, 12th March, 7.15 p.m. General Meeting in the Town Hall Assembly Room. Paper : "The Development of the De Havilland Series of Light Aircraft Engines", by J. L. P. Brodie, M.I.Mech.E.

NORTH-EASTERN
Wednesday, 21st March, 7.30 p.m. General Meeting in the University Chemistry Lecture Theatre, Leeds. Paper : "The Development of the De Havilland Series of Light Aircraft Engines", by J. L. P. Brodie, M.I.Mech.E.

NORTH-WESTERN
Wednesday, 14th March, 7 p.m. General Meeting in the Grosvenor Hotel, Chester. Paper : "Wear of Fuel Injection Equipment and Filtration of Fuel for C.I. Engines", by A. E. W. Austen, B.Sc., Ph.D., A.M.I.Mech.E., and B. E. Goodridge, A.M.I.E.E.

SCOTTISH
Monday, 19th March, 7.30 p.m. General Meeting in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper : "The Development of the De Havilland Series of Light Aircraft Engines", by J. L. P. Brodie, M.I.Mech.E.

WESTERN
Thursday, 29th March, 6.45 p.m., in the Royal Hotel, Bristol. Paper : "The Development of the De Havilland Series of Light Aircraft Engines", by J. L. P. Brodie, M.I.Mech.E.

The following meetings will be held during April :—

LONDON
Tuesday, 10th April, 5.30 p.m. General Meeting at Storey's Gate, St. James's Park, S.W.1. Paper : "The Development of the De Havilland Series of Light Aircraft Engines", by J. L. P. Brodie, M.I.Mech.E.

NORTH-WESTERN
Wednesday, 4th April, 7.15 p.m. General Meeting in the Walker Engineering Laboratories, The University, Liverpool. Paper : "The Performance and Weight of Automobile Petrol Engines", by Donald Bastow, B.Sc. (Eng.), M.I.Mech.E. (Member of A.D. Council).

WESTERN
Thursday, 12th April, 6.45 p.m., in the Park Hotel, Cardiff. Paper: "Some Factors Governing the Performance of Crankcase Lubricating Oils", by A. Towle, M.Sc. (Eng.), M.I.Mech.E.

STRESS ANALYSIS OF CYLINDER BLOCKS

THE use of wire resistance strain gauges in determining the mechanical properties of cast iron is demonstrated and an example of the stress analysis of a casting is given by M. A. Erickson in *A.S.T.M. Symposium on Testing of Cast Iron with SR-4 Type of Gauge.*

Results of tests of mechanical properties are given as stress-strain curves for both tension and compression. The specimens were first loaded step-by-step, a return to zero load being made between each step, and in subsequent tests were loaded continuously. These tests gave results which are referred to as elastic stress-strain curves and from such curves of axial and circumferential strains Poisson's ratio was calculated. The permanent set in the step-by-step tests is also given. From the results it is concluded that, for the cast iron used, stress is not proportional to strain and plastic strain occurs immediately upon the initial load application, becoming greater at an increasing rate until failure occurs. Moreover, the plastic strain produced by tensile loading is removed by compressive loading of approximately equal intensity. Poisson's ratio varied from 0.27 at 4,000 lb. per sq. in. to 0.18 at 40,000 lb. per sq. in. From the foregoing, it becomes apparent that the stress calculated from a gauge reading on a casting may be erratic unless the values of the modulus of elasticity and Poisson's ratio are obtained from the stress-strain curve at a value of strain corresponding to the strain under consideration in the part.

The example given of a failure problem in an iron casting is that of an automotive engine block which failed in the outer wall. Stresses were first measured due to cylinder head stud tightening and engine operation, and then the residual stresses were measured by the relaxation method. The readings were analysed for maximum and minimum strains and the stresses computed from the elastic curve as found from the specimen test. The analysis indicated that a residual tensile stress of 26,000 lb. per sq. in. existed on the inside of the casting wall and this, when combined with the assembly and operational stresses and interpreted using the Goodman diagram, gave a low factor of safety. The casting was made stronger by thickening the section. This may not have altered the residual stresses appreciably but would increase the factor of safety by lowering the assembly and operating stresses.

[M.I.R.A. Abstract No. 5119]

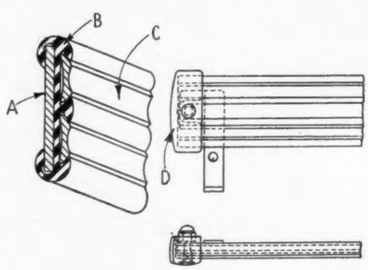
CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Decorative Moulding for Bumpers

A PROTECTIVE and decorative moulding of natural or synthetic rubber cr plastic material is applied to a flat strip bumper bar. It is particularly, but not exclusively, suitable for the bumpers and guards of public service vehicles. The bumper bar itself may be of steel or of a suitable light alloy if it is desirable to keep down the total weight. Fitted to the front face of the bar A, the moulding B has return margins which embrace the upper and lower edges of the bar and bear closely on the rear face. Upper and lower edges of the moulding are rounded and given a smooth finish to facilitate cleaning.

In the front face of the moulding is formed one or more channels with inwardly directed flanges and into these are fitted rolled, extruded or sheet metal filler strips C of light alloy. Each end of the assembly is enclosed in a hollow die-cast or pressed



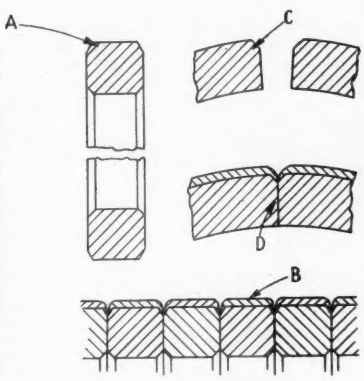
No. 634280

end cap D secured by a bolt passing through the bar.

The bars may be straight or curved as required, and two bars may be mounted in superimposed relationship on the same attachment brackets. *Patent No.* 634280. *Weathershields, Ltd., and W. H. Bishop.*

Chromium Plated Piston Rings

PREPARATORY to plating the peripheral surfaces of piston rings, it is customary to assemble a number of rings on a mandrel and clamp them between end plates to form a cylinder. The chromium is electrodeposited as a uniform layer over the whole outer surface of this cylinder. In subsequently separating the rings the chromium layer must be broken which results in either jagged edges or local peeling of the plating. It is proposed, therefore, that at the corners where the side faces meet the peripheral surface of the ring, a chamfer or a radius A be provided. Preferably, this is a chamfer at an angle of about 30 deg. to the side face and extending approximately 0.015 in down the side face. When rings thus modified are assembled on the mandrel they form a peripherally grooved cylinder and the deposit B of chromium will taper in thickness from the mouth to the bottom of the grooves. Consequently, the chromium layer is smoothly rounded off at the corners of the rings and is relatively very thin at the bottom of the grooves so that the rings can be easily separated to leave clean edges.



No. 633457

In like manner a chamfer C is provided at the gap ends of the ring. The ends may be abutted, as shown at D, or closed against a thin plastic insertion, as is more usual. It is assumed that the rings will be mounted on the mandrel with their gaps in alignment.

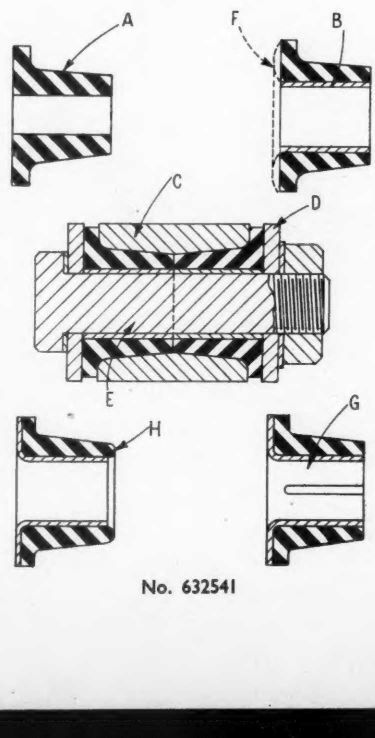
The method can also be adopted for rings furnished with a layer of a softer material such as tin, cadmium, zinc or lead. *Patent No.* 633457. *Hepworth and Grandage Ltd.*

Flexible Bearings

IN a separately assembled flexible bearing a pair of resilient bushings, each stretched over a cylindrical inner member, are axially compressed to produce a radial deformation which ensures a firm frictional grip in the outer member. Each flanged bushing A is formed with a cylindrical bore and a frusto-conical outer surface and when stretched over a metallic inner member B its initial length and radial thickness is appreciably reduced. The outer member C has a double frusto-conical bore of slightly smaller mean diameter so that when the bushes are assembled between side plates D and are drawn up by a clamp bolt E they will firmly adhere to both inner and outer members and also provide satisfactory resistance to axial displacement.

A modification aimed at securing a greater value of axial compression and at the same time reducing the risk of the bushing material being pinched between the side plate and inner member B is secured by the provision of a convex face to the flange, as indicated in broken outline at F.

In an alternative construction, a flanged inner member G may be used, and if necessary the cylindrical body of this member may be slotted to permit the radial pressure exerted by the bushing to press it into close contact with the clamp bolt. As the flanged inner member G precludes the use of a convex face on the flange of the bushing extra compression may be obtained by a convex extension of the inner face of the bushing, as at H. *Patent No.* 632541. *Silentbloc Ltd., and V. A. Trier.*



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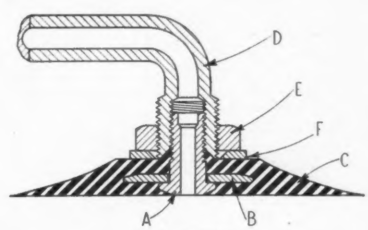


No. 633092

Interchangable Tyre Valve

DIFFERENT vehicles have wheels calling for different types of tyre valves. It is common practice for valves to be permanently secured to the tube and consequently stocks of similar tubes having different types of valves must be built up and stored against demand. The proposal is to fit the tube with a standard valve foot to which specific valve bodies may be fitted as required.

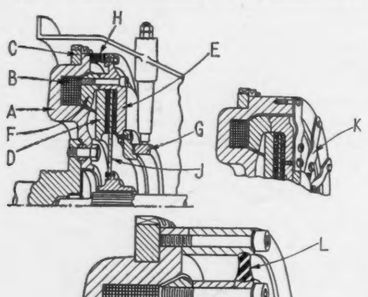
The tubular foot A and its anchor ring B are embedded in the moulded rubber base C which is bonded to the inner tube. Externally, the foot is threaded to receive the internally threaded end of the valve stem member D. The inner end of the stem is coned to co-operate with a conical seating formed on the base C. Due to the inherent resilience of the rubber seating, the angular position of the stem may be adjusted without impairing the air-tightness of the junction. A nut E screwed on the stem and gripping the base through an interposed clamping washer F secures the stem in its adjusted position.

Another specification, No. 633357 by the same Patentees, shows a different construction to achieve the same ends. *Patent No.* 633092. *Manufacture de Caoutchouc Michelin* (*France*).

Magnetically Operated Frictional Clutch

THIS invention concerns the provision of a torque-transmitting element between the two engaging members which will permit the necessary axial movement without any tendency to stick or jam. One member comprises a flywheel A carrying the magnet coils B, starter ring C and a rigidly attached clutch ring D. Presser plate E, carrying the armature F and the slip-rings G, is spaced from the flywheel by a flexible metal bellows H. Between the flywheel clutch face and the presser plate is located a conventional clutch disc J mounted on a hub splined on the driven shaft. The bellows which transmits torque to the presser plate is given a spring bias to ensure disengagement when the magnets are de-energized.

In a modified construction the bellows is replaced by a series of radially positioned strips K of spring steel attached by screws to the flywheel flange and the presser plate. Another alternative is to use a rubber annulus L permanently

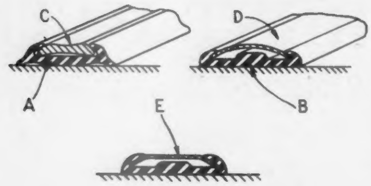


No. 633632

bonded to the presser plate and a detachable flywheel flange. It will be noted that the rubber annulus tapers in thickness in order to provide a bonding area on the inner peripheral surface at least equal to that of the outer peripheral surface of greater diameter. *Patent No.* 633632. *Humber Ltd., A. C. Miller and A. M. Kamper.*

Trim Moulding

THIS moulding to conceal bodywork joints or to impart an ornamental finish is of composite construction with a



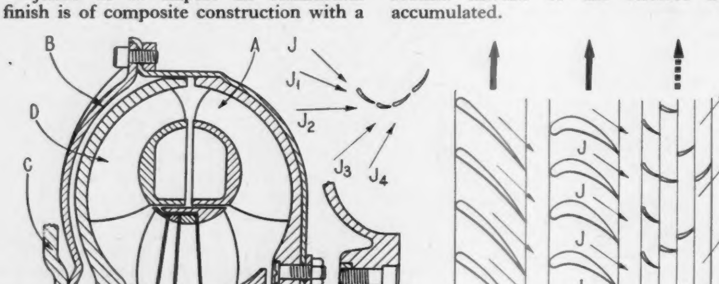
No. 634256

resilient moulded base fitted with a metal insert. In all the embodiments illustrated the base strip A is a moulding of natural or synthetic rubber or a plastic material of a channel section with inwardly directed flanges or lips. The bottom surface may be either flat or initially concave so that when secured to the body structure the tapered edges of the base are pressed into close contact with the panel surface. If desired, the base strip may be moulded with a central rib B of substantial thickness on its upper surface to receive the nails or screws by which it is attached to the body structure. Filler strips of aluminium or light alloy may be polished, anodized or painted to match or contrast with the general colour scheme of a vehicle. It may be a rolled or extruded strip, as at C, with a concave, flat or convex exposed face. Alternatively, the insert may be a sheet metal strip D of convex form or a flat strip E with inclined edges. *Patent No.* 634256. *Weathershields, Ltd., and W. H. Bishop.*

Hydraulic Torque Converter

BY the use of a reaction member divided into four sections, individually mounted on overrunning clutches, it is suggested that this converter will meet the requirements of an automobile transmission without the need for additional mechanical gearing.

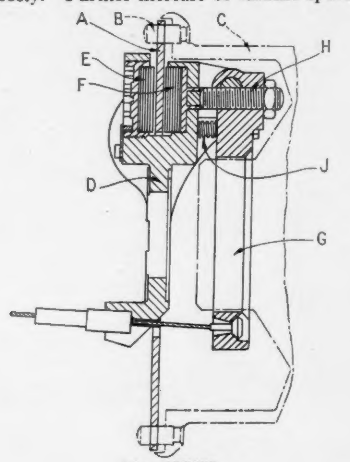
Impeller A is mounted in the casing B attached to flange C secured to the engine crankshaft or flywheel. Turbine D is bolted to a hub E splined to output shaft F. The fluid circuit is completed by reaction sections G$_1$, G$_2$, G$_3$, G$_4$, each mounted on a sprag-type overrunning clutch on a common sleeve H encircling the output shaft and splined at the rear to a hub member bolted to the transmission housing. This method of furnishing a clutch between each reaction section and the sleeve instead of between adjacent sections, with a final clutch between the last section and the fixed member, ensures that each clutch carries only the load of its particular section instead of the stresses being accumulated.



No. 633161

The reaction blades are of airfoil section to minimize disturbance to flow. Each section has a different number of blades, preferably a prime number, to avoid setting up harmonic vibrations. In the flow diagram the impeller A and turbine D rotate in the direction indicated while the reaction sections are free to rotate in the same direction but are restrained from rotation in the opposite direction. Arrows J indicate the path of the fluid as determined solely by blade angle and curvature. Under starting conditions, with the turbine stationary, fluid impinges on the faces of all the reaction sections so all co-operate to form a complete reaction member. The torque ratio is then in excess of 3 : 1.

As the turbine begins to rotate, however, a circumferential movement is superimposed on the component of flow from between the blades and, at the resultant angle J$_1$, the fluid impinges on the back of reaction section G$_1$, which then coasts freely. Further increase of turbine speed



No. 635603

progressively changes the flow angle to J$_2$, J$_3$, J$_4$, until all sections run freely and the unit functions as a fluid coupling with a torque ratio of 1 : 1. *Patent No.* 633161. *Ford Motor Co., Ltd.*

Disc Brake

IN this aircraft-type brake the annular plate A floats on pins B mounted near the rim of the drum or hub extension C. The brake plate D is forked to support a pair of friction pads on opposite sides of plate A. The fixed pad E is adjustably mounted in a screwed housing while the axially movable pad F is carried on a plunger, the stem of which slides through a bore in the inner limb of the fork.

On lugs extending from the brake plate is pivotally mounted a pressure lever G which is either bowed or looped to clear the protruding boss of the wheel hub. An adjustable abutment screw H, mounted near the pivot axis to obtain maximum leverage, abuts the end of the stem of friction pad F. A pair of helical springs J, carried on pins screwed into the brake plate, serve to retract the lever when braking pressure is released. Only one pair of pads is shown but two or more pairs may be used with suitably modified actuating levers.

In the example, operation is by means of a flexible cable, but the specification also shows applications arranged for hydraulic or pneumatic operation. *Patent No.* 635603. *The India-Rubber, Gutta Percha and Telegraph Works Co., Ltd. F. J. Tarris and D. Webb.*